PRODUCT ENGINEERING

MAY, 1946

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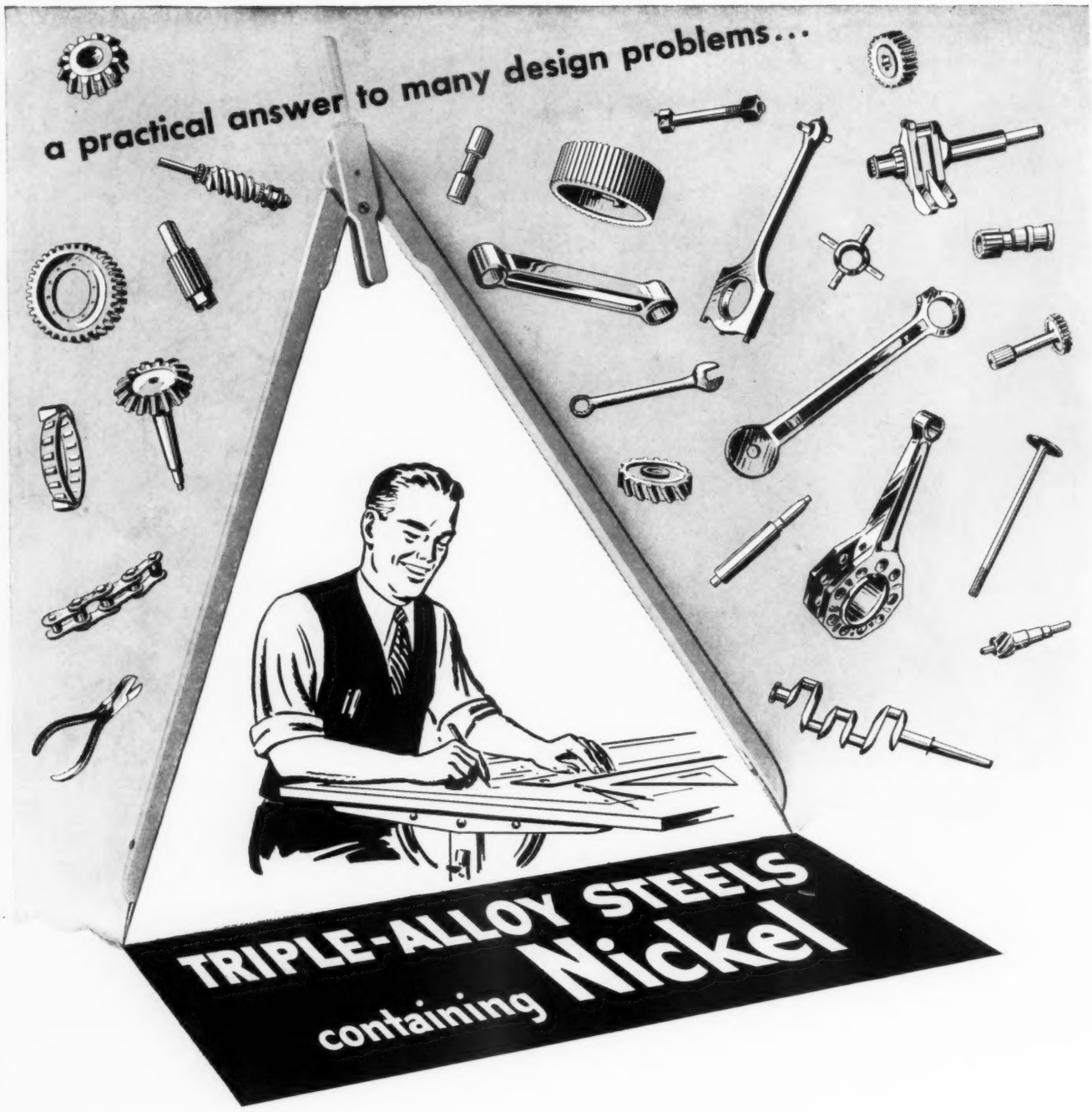

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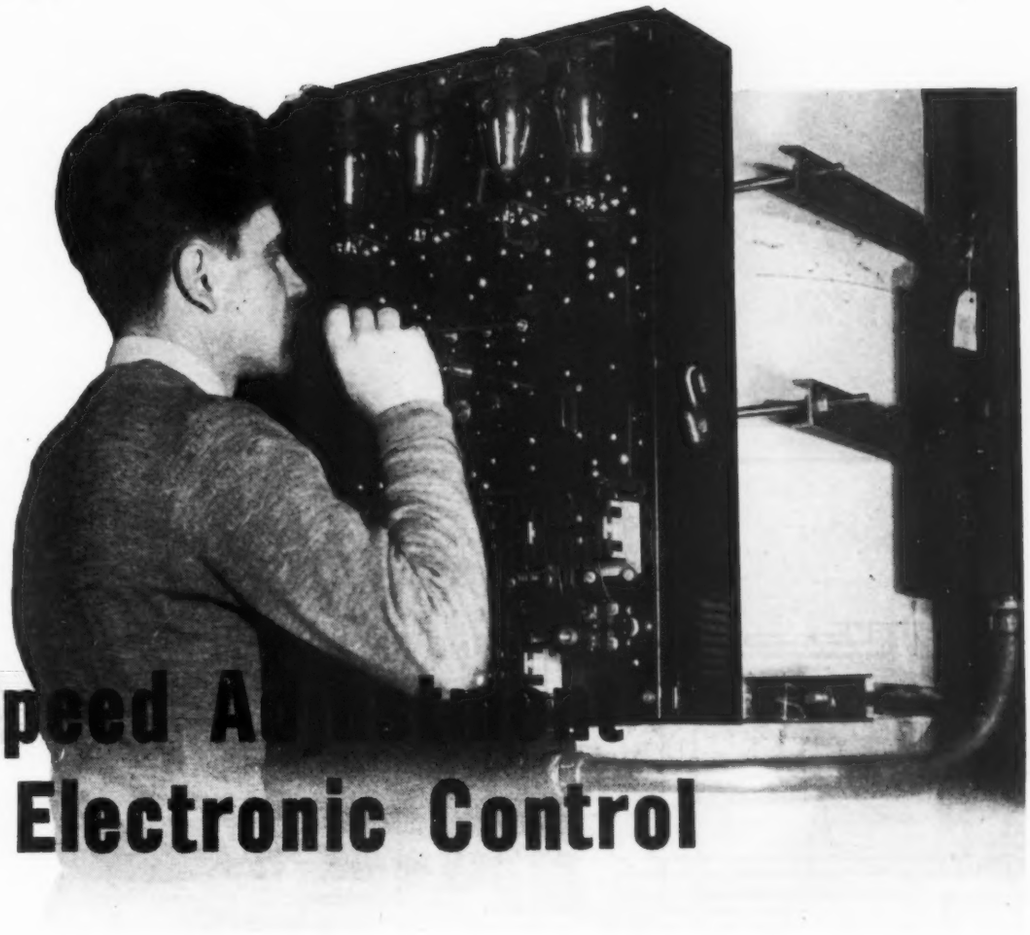


Motor Speed A[illegible] By Electronic Control

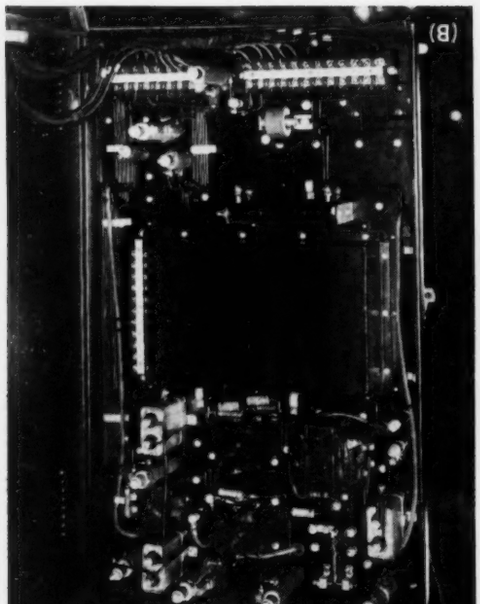
S. S. WOLFF
Chief Engineer, Century Electric Company

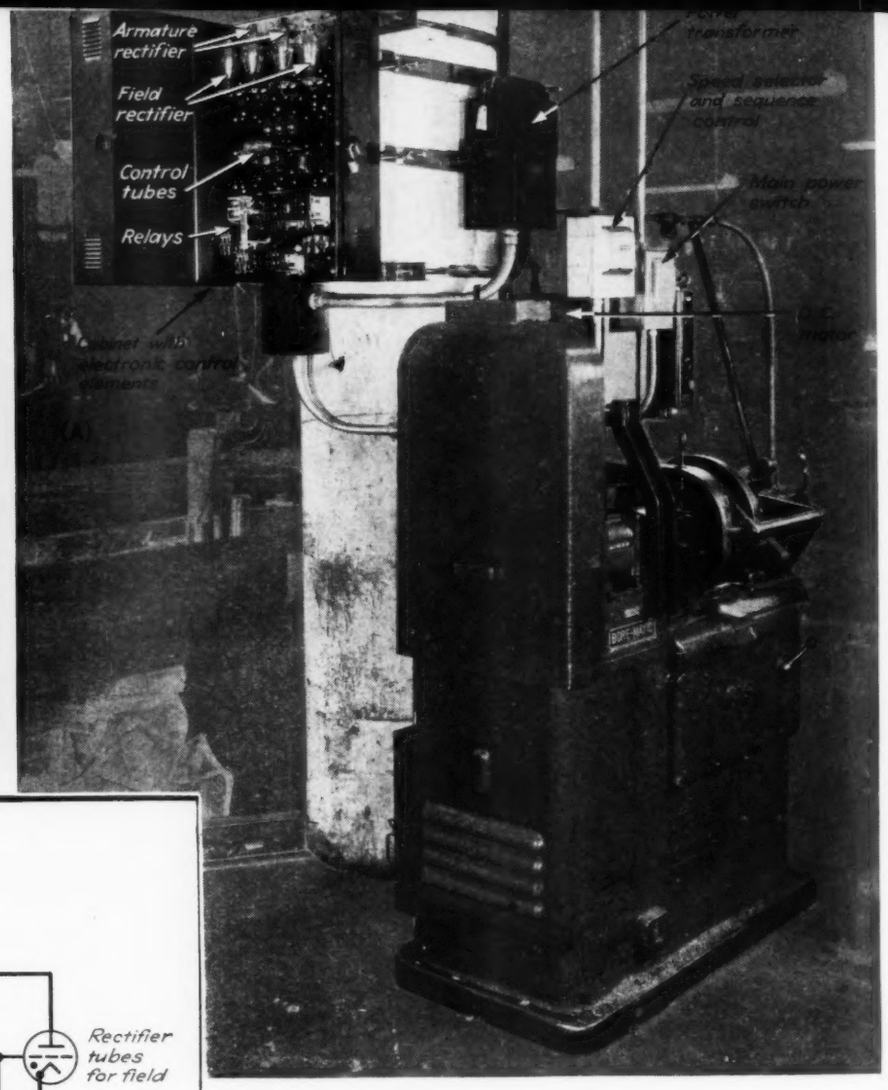
A progress report on electronic control based primarily on recent engineering tests of a 1 hp. unit. Performance characteristics, motor heating, and noise are typical of the factors discussed on a relative basis. Oscillograms show current and voltage conditions existing in the motor for different load conditions at 60, 1,200, and 3,600 r.p.m.

SECURING THE ACCEPTANCE of electronic control by industry during the next few years will not be a simple task, despite the many successful and fascinating accomplishments of electronics during the war. Introducing any fundamentally new equipment into the general industrial market is always difficult, and this is especially true of a technical development like electronic motor control that must be applied with care. Engineers who are considering whether or not to apply electronic motor control to their products may find the discussion that follows later in this article helpful in evaluating relative factors such as size, heating, efficiency, and noise.

Like most new developments, electronic controls for industry have been overpublicized and to a degree this is true of electronic motor control. The real problem of product designers is to determine just when to use electronic motor control so that the improvement in overall performance warrants its use. Electronic control may add to the sales appeal of a product or machine, but in addition the user must gain operating advantages that are greater than any disadvantages introduced. Manufacturers that attempt to create a new demand for an existing machine by the addition of electronic control must be careful that such a drive is economically sound to prevent doing themselves more harm than good.

Ever since the introduction of alternating current as the source of power for industrial drives there has been a demand for an a.c. system that provides the flexible speed control of the d.c. motor. The most generally used method is the variable voltage system, with an a.c. to d.c. motor generator set supplying power to a d.c. motor. With this system, constant voltage is supplied to the field circuit of the motor and adjustable voltage to the armature circuit. The degree of speed





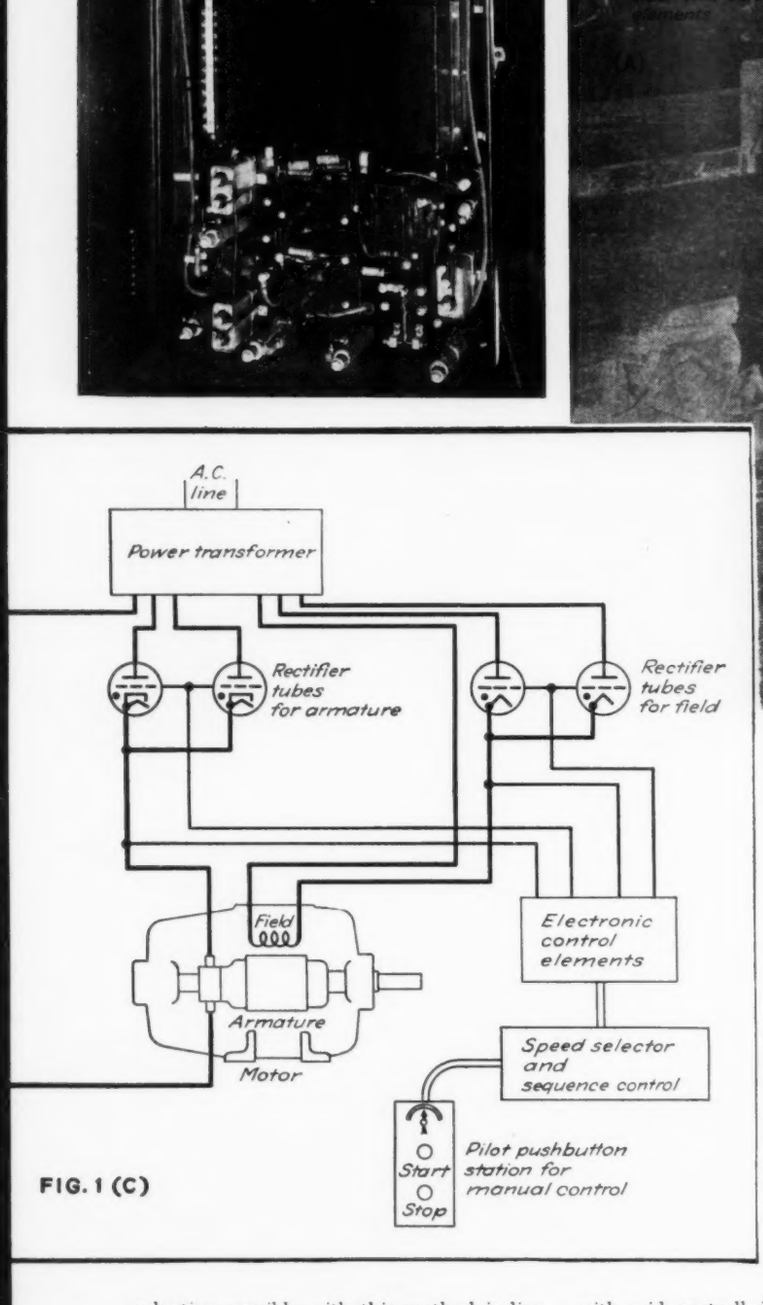


FIG. 1 (C)

Fig. 1—Production was substantially increased on this borizer by replacing the single speed a.c. driving motor by electronic motor control. Previously, castings were machined at the same speed regardless of their size, but electronic control now drives the borizer at the best speed for the particular size of casting being machined. (A) The drive for this boring machine essentially consists of a d.c. motor, a cabinet with panel-mounted rectifiers and control tubes, an anode transformer, and a control station. (B) Many of the components of the electronic control are mounted on the back of the panel, which swings out from the cabinet for easy inspection and maintenance. (C) Simplified schematic of the electronic motor control. A throwover switch in the selector and sequence control unit makes it possible to operate the borizer manually from the pilot pushbutton station or automatically by limit switches on the machine tool. At predetermined intervals, the limit switches control the grid bias on the rectifier tubes supplying the armature and field in such a manner that the speed of the spindle changes with the diameter of the part being machined.

reduction possible with this method is limited usually by the residual voltage of the generator. Where extremely low values of voltage are required, such as in passenger elevator operation, a tapped potentiometer method of generator field control can be used.

The electronic control system is similar to the motor generator set except that the conversion from a.c. to d.c. is done by a rectifier tube. The conventional electronic control drive includes a d.c. motor, a panel with grid controlled thyratron rectifiers and auxiliary control tubes, an anode transformer, and a control station, as shown by Fig. 1. Since the power rectifiers for the armature and field circuits are separate they can be varied independently of each other. Electronic control can supply the motor with fixed field voltage and adjustable armature voltage; or for greater range of speed, with both adjustable armature and field voltage.

Although a motor generator set is more sturdy and withstands overloading better than electronic tubes, it does not have the practically instantaneous speed of response of vacuum tubes. In addition, a high degree of sensitivity can be obtained by galvanometer operated relays; and with the wide variety of tubes and circuit elements available endless combinations can be used to change the motor characteristics by remote control. These advantages plus the resulting stepless speed control make the electronic system often more desirable.

However, against these advantages should

be balanced other factors such as cost, maintenance, and certain technical limitations.

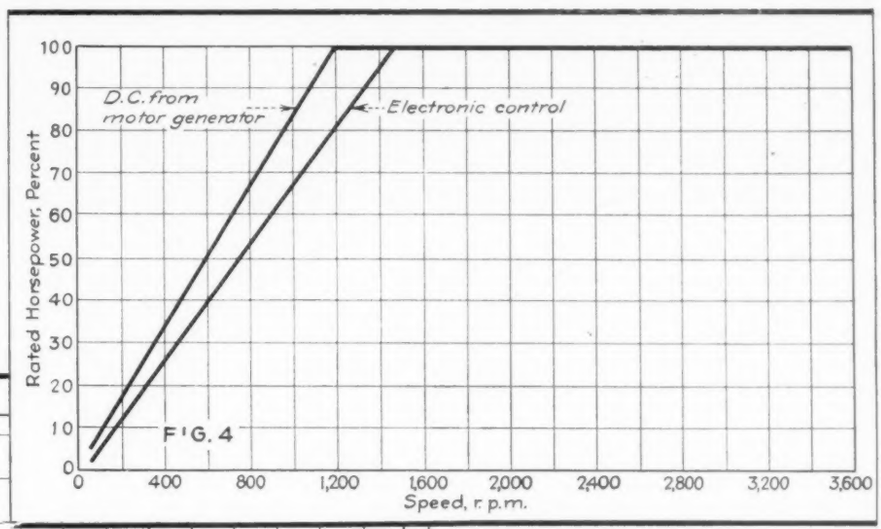
Compared to a motor generator set with necessary control, the first cost of the electronic control is 15 to 50 percent more, depending upon the features required. Likewise the d.c. motor used with electronic control is larger and more expensive than a motor of the same horsepower but supplied from a motor generator set. Not only first cost, but upkeep of the electronic control is usually higher. The maintenance of a motor generator set involves lubrication, replacement of brushes, commutator cleaning and trueing, and similar well understood tasks that any good mechanic can do. However, the complexity of the electronic control circuits requires the service of a specially trained technician.

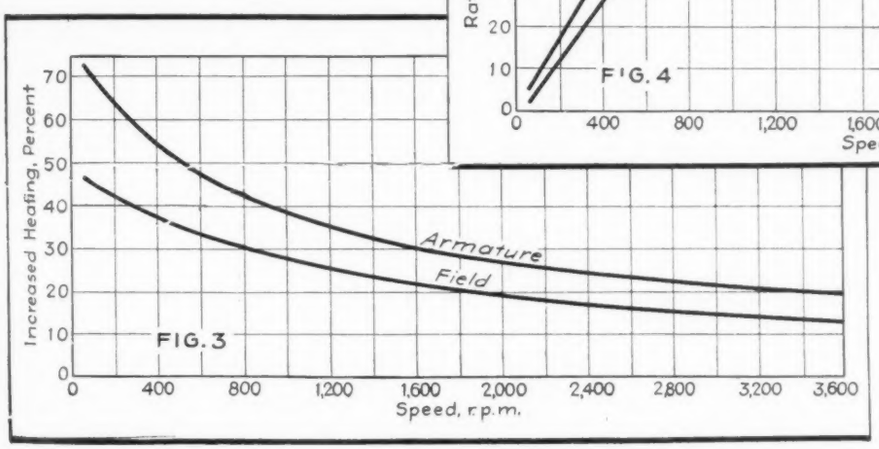
The following explanation and analysis of test data on a 1 hp. motor with constant torque characteristics from 1,200 to 60 r.p.m. and constant horsepower characteristics from 1,200 to 3,600 r.p.m., may clarify some engineering factors that may be helpful when considering the use of electronic motor control. Fig. 2 shows some of the equipment and connections used in making the tests.

At low speeds, the heating of the electronically controlled motor is increased considerably because the d.c. supplied by the rectifier tubes is pulsating, not constant. Also, ventilation is reduced. Fig. 3 shows the increase in heating of the armature and fields of a d.c. motor when operated by electronic control as a percentage of the corresponding heating at the same horsepower outputs when supplied with d.c. from a motor generator set. These data are for single phase full wave rectification which is the most economical for a 1 hp. unit. On larger motor applications that can economically use polyphase rectification, the difference in heating is not as great as that shown in Fig. 3 and the relative cost of the motors is correspondingly reduced.

Within the permissible temperature limits, Fig. 4 shows the horsepower output of a d.c. motor when operated by rectified a.c. as a percentage of the horsepower available when the same motor is operated by d.c. from a motor generator set.

The wave shapes of the rectified a.c. supplied to the armature and field circuits at





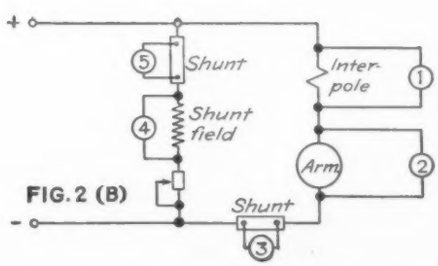
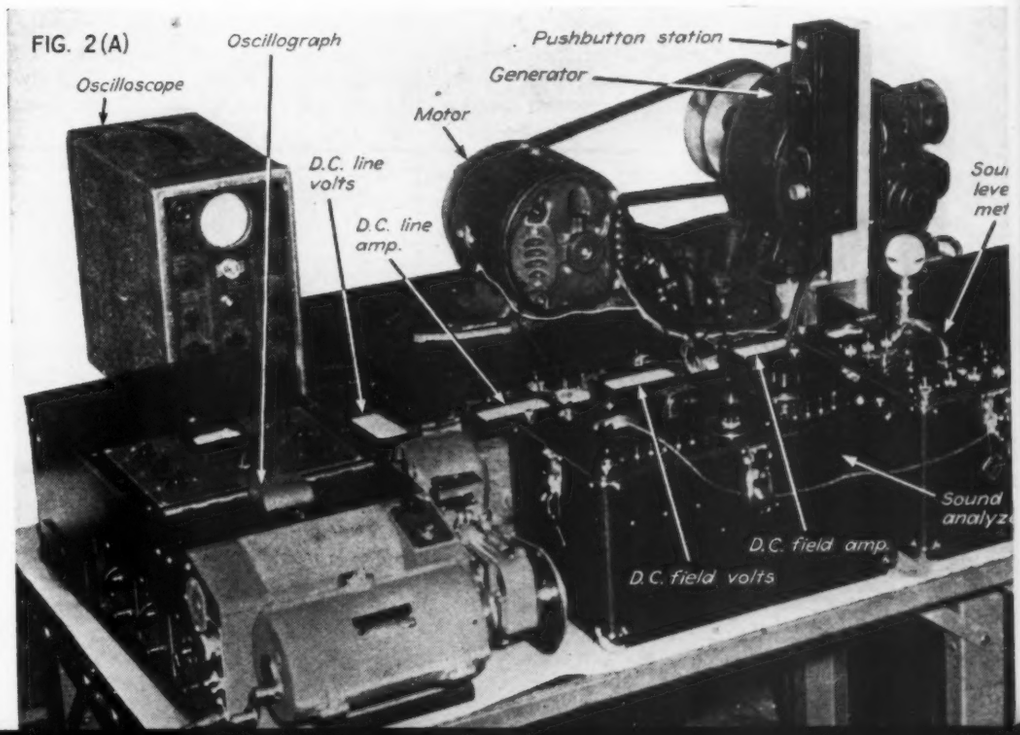


Fig. 2—Wave shape, voltage, and current conditions of a 1 hp. d.c. shunt motor when operated on rectified a.c. at different speeds were obtained by test. (A) View of the testing equipment; a.c. values were read from switchboard instruments and are not shown in photograph. (B) Values of voltage or current were recorded by the oscillograph at the five points shown.

Fig. 3—Relative increase in armature and field heating when motor is operated by electronic control as compared with operation from d.c. power.

Fig. 4—Curves comparing motor output on the basis of permissible temperature rise when operated on d.c. and by electronic control.



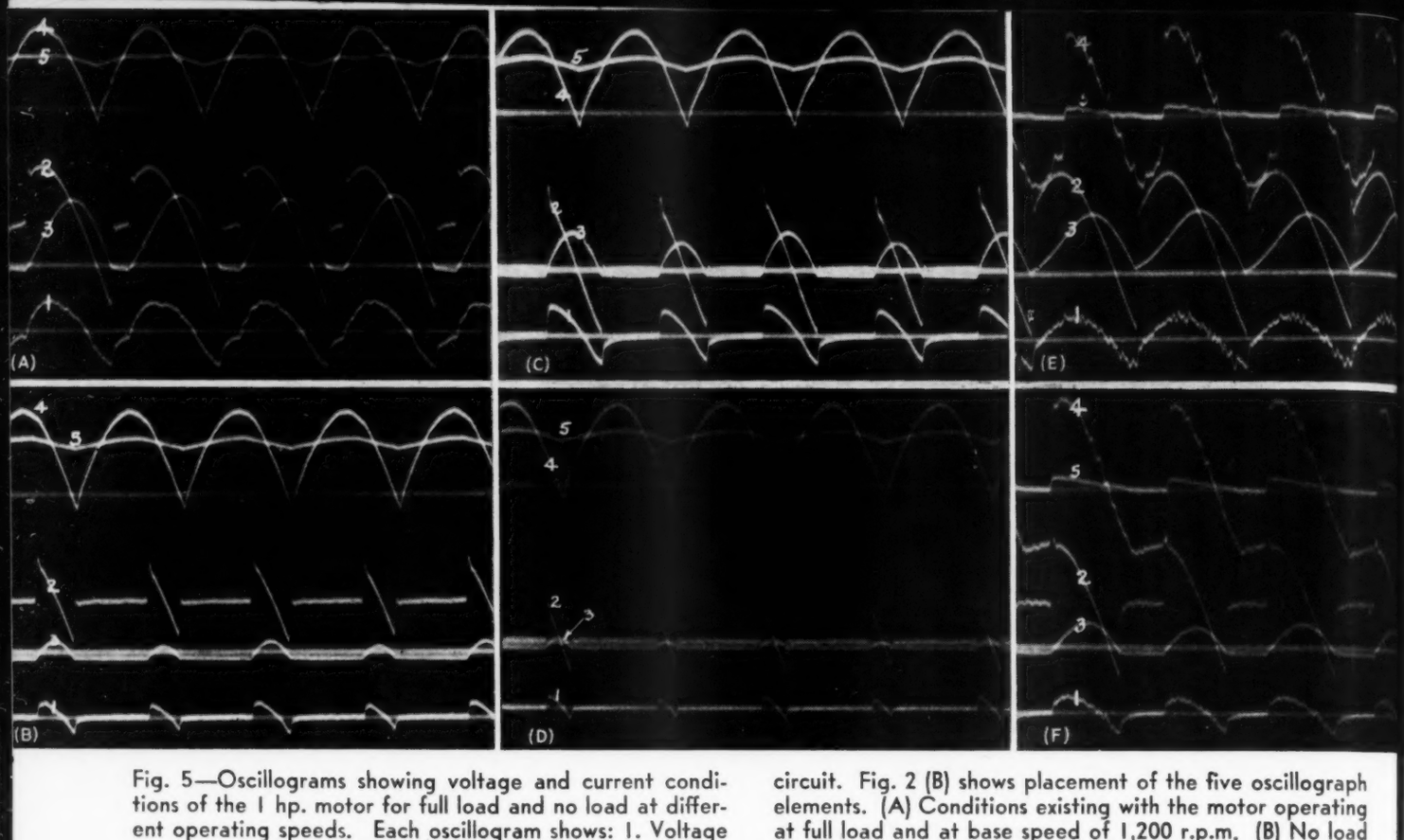


Fig. 5—Oscillograms showing voltage and current conditions of the 1 hp. motor for full load and no load at different operating speeds. Each oscillogram shows: 1. Voltage across the interpole winding; 2. voltage across armature circuit; 3. current flowing in armature circuit; 4. voltage across the shunt field; and, 5. current flowing in shunt field circuit. Fig. 2 (B) shows placement of the five oscillograph elements. (A) Conditions existing with the motor operating at full load and at base speed of 1,200 r.p.m. (B) No load and 1,200 r.p.m. (C) Full load (constant torque) at 60 r.p.m. (D) No load at 60 r.p.m. (E) Full load (constant horsepower) at 3,600 r.p.m. (F) No load at 3,600 r.p.m.

various speed settings are shown in Fig. 5 by the various oscillograms.

Where heavy sustained overloads are encountered the motor generator set has advantages over the electronic control. Since the rectifier tubes are relatively expensive they are selected and applied to carry the full load current of the motor on a continuous duty cycle basis. Although the tubes are capable of carrying a peak current of about 200 percent of rated current for short periods of time, if this overload continues for even one-third of a minute serious overheating of the tubes results. To prevent dangerous overheating the grid voltage automatically controls the maximum current that the motor can draw.

Electronic control has an advantage over the motor generator set because within the full load limits of the system closer speed regulation is possible. This is done by compensating for the resistance drop in the armature circuit and results in about two percent speed regulation from full load to no load at the 1,200 r.p.m. base speed. This regulation increases as the motor speed is decreased but the control provides adjustment within certain limits. Other features that can be incorporated in the electronic control include dynamic braking, reversing service, overload and low voltage protection.

Operating cost of the electronic unit is less because its overall efficiency is higher than that of the motor generator set, as shown by Fig. 6. Electronic controls using polyphase rectification show even greater relative savings. The power factor of the motor generator set is better than that of the electronic unit as shown by Fig. 7. Polyphase rectification would give about the same relative values because these curves are for a single-phase repulsion-induction motor with inherently low power factor, and in making a comparison with polyphase rectification the power factor for a polyphase squirrel cage motor should be used.

It is difficult to make a comparative analysis of the cost of maintenance of the two systems. In electronic control the cost of tubes, condensers, and resistors is quite high and their life is variable. To assure long life of the tubes, the applied voltage should be maintained within plus or minus five percent of rated voltage. With motor generator sets the voltage variation can be plus or minus ten percent without impairing the operation of the unit, and even greater variations can be tolerated intermittently without serious difficulty. As progress is made in the art, tubes with longer life and of more en-

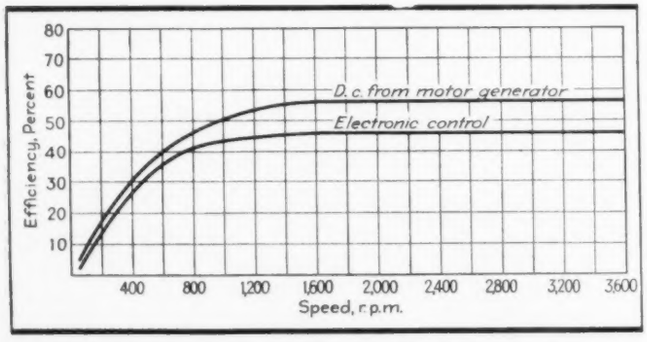


Fig. 6—Curves comparing the overall efficiency of a 1 hp. motor operated by electronic control with the same motor operated on d.c. supplied from a motor generator set.

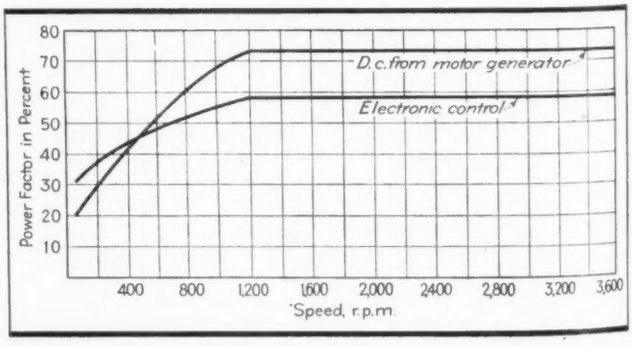


Fig. 7—Curves comparing the power factor of a 1 hp. motor operated by electronic control with the same motor supplied with d.c. by a single-phase motor generator set.

during construction probably will be developed.

Motor noise is another factor that may be important when considering electronic control. A great many studies have been made and much written on the physiological and psychological effect of noise on the efficiency of workers. As a result, purchasers of electrical machinery are becoming increasingly noise conscious and manufacturers of electric motors are making every effort to reduce noise to a minimum. Quietness of operation is rapidly becoming a requisite of good motor design.

The human ear is sensitive to sound energy ranging in frequency from approximately 60 to 8,000 cycles per second. Variations in intensity depend upon the individual, his age, physical condition and similar factors. Although the normal ear does not respond equally to all frequencies, it is most sensitive to values in the proximity of 2,000 c.p.s. and becomes less sensitive at both lower and higher values.

In conducting tests on electronically controlled motors it was observed that noise and vibration were appreciably greater when operating on rectified a.c. than when the same motor was operated on d.c. power. Although the difference in noise levels was not very great when both the armature and field were excited at rated voltage, there was an advantage in favor of d.c. operation. This advantage diminished as the field was weakened while the armature voltage was held at its rated value. As the field strength was reduced, the speed of the motor increased until the ventilating fan noise became the predominating factor.

To eliminate any noise emanating from the loading devices, sound measurements also were made on the motor when operating at no load. When operating at no load with rated voltage on the field and reduced voltage on the armature, d.c. showed a definite advantage. At the minimum speed at 60 r.p.m., or 1/20 of the 1,200 r.p.m. base speed, the sound pressure on rectified a.c. was approximately five times as great as on d.c. power. On some applications this noise would be definitely objectionable. On other applications, where the general noise level is already high and little importance is attached to the separate elements, the motor would be acceptable.

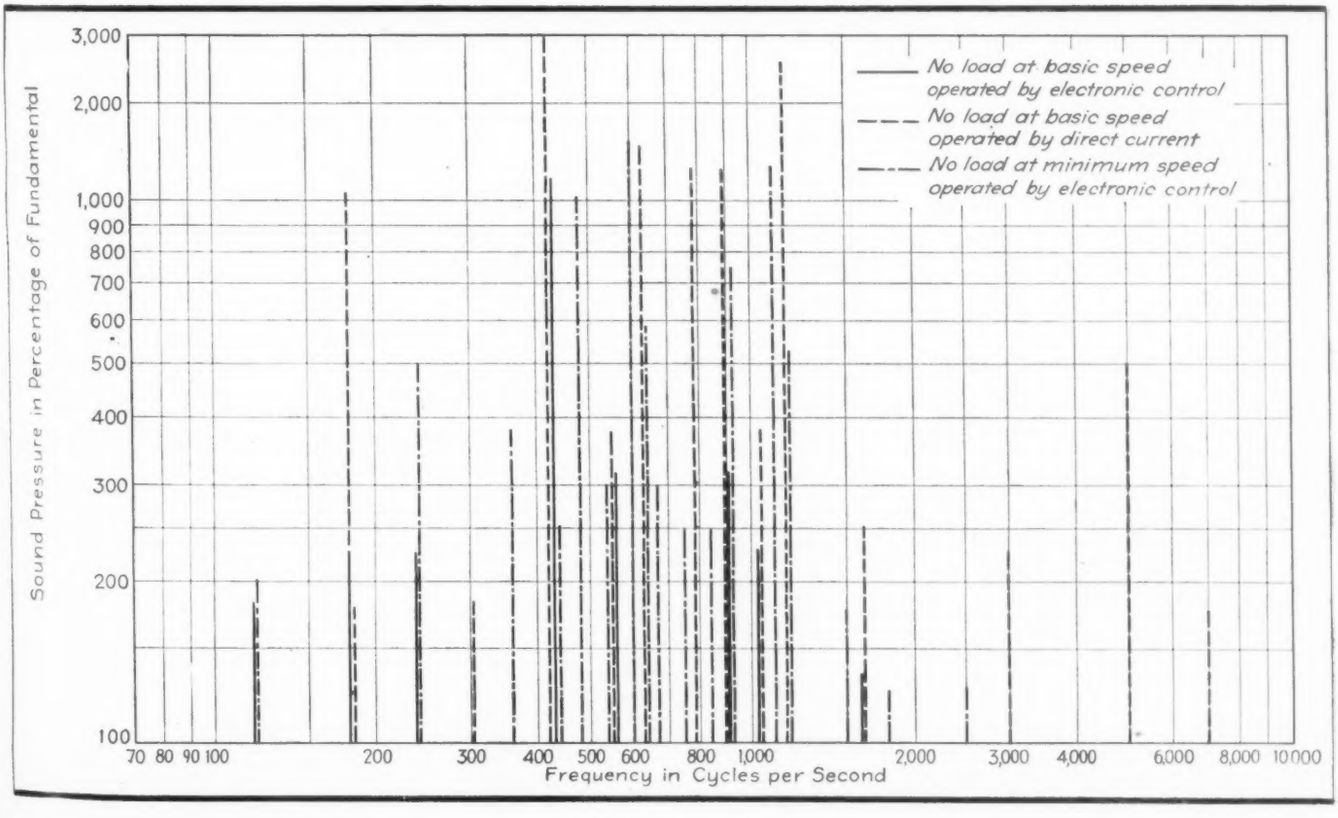
The sound spectrum of the 1 hp. motor when operating on rectified a.c. and when using a motor generator set as the source of d.c. is shown in Fig. 8. Three conditions are illustrated: First, with the motor operating on rectified pulsating d.c. at 1,200 r.p.m. with no load applied; second, with the motor operating under similar conditions but supplied with continuous d.c.; third, with the motor running on rectified pulsating d.c. at 60 r.p.m. and no load.

An analysis of this kind is helpful in determining which harmonics of a complex sound wave are strongest and must be reduced to quiet a machine. When this motor was operated from continuous d.c. at 60 r.p.m. and without load, no noise could be measured above the level of the room and hence no analysis could be made. In all of the above tests the surrounding noise level was at least the customary ten decibels below the measured noise. These analyses and the wave shapes shown in Fig. 5 indicate the effect of electrical design and the wave shape of the power supply on motor noise.

The factors producing noise are also instrumental in producing vibration. When a motor is to be mounted directly on a machine tool where vibrations cannot be tolerated the motor must be designed with this in mind, or a mounting must be used to absorb the vibrations. As the range of speeds of any particular motor increases this problem becomes more acute.

Although there has been considerable research work done with electronic control of motors relatively little has been published. The reason probably is not because the information is a trade secret, but because engineers are reluctant to disclose results of experiments until sure that findings of today will not be obsoleted by the progress of tomorrow. However, perhaps the electronic control industry should not wait for perfection, but instead by progress reports expose the path the development is taking. Like all new developments, electronic motor control will eventually find its economic place in industry provided engineers will continue to apply it only where it pays its own way.

Fig. 8—Sound spectrum of a 1 hp. motor for different conditions of load and speed comparing d.c. operation with electronic motor control.



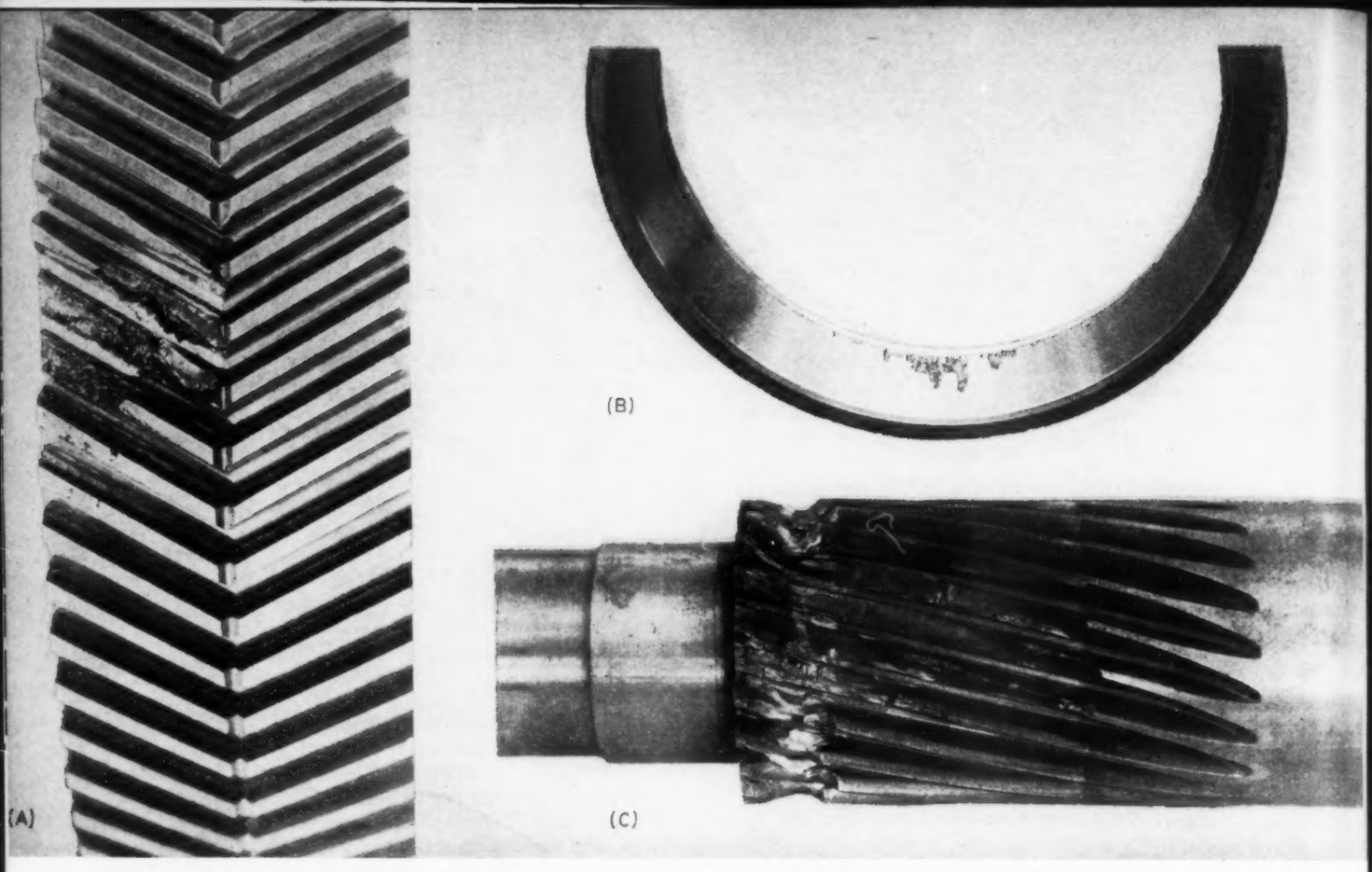


Fig. 1—*(A)* Sub-surface defects, which occur in cast materials, can be detected by radiographic inspection; most of them can be avoided by good design and foundry practice; *(B)* Failures of bearing races as the result of out-of-round housings often can be prevented by relieving the bearing bores at the parting lines of spit housings; *(C)* Excessive deflections and failure by the designer adequately to specify the heat treatment were responsible for the failure illustrated by this opinion.

Practical Considerations That Affect Gear Durability in Speed Reducers

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Principles and practices underlying the design of gears for speed reducers, including a discussion of the factors that influence gear life. Practical methods of evaluating these factors in design use are presented. Typical failures of gears are illustrated and analyzed.

THE DESIGN of geared speed reducers is a combination of science and experience. Because of the complex and variable nature of the loads and stresses influencing the performance of gearing, all the factors that affect gear durability cannot be included in a standard gear life formula or design code. Experience must be relied upon for the evaluation of some of the factors. For this reason, conventional formulas for the design of gears apply only to tooth durability and tooth proportions.

Other factors that influence the life of a gear train are the gear blank, its web design, mounting on shaft, housing and shafting deflections, housing cleanliness, out-of-round bearings, torsional deflections, defective heat treatment, area of tooth contact and others. No way has been discovered to determine the influence of all of these factors separately, but an evaluation of them can be made thorough the study of gear performance under actual service conditions.

Most designers neglect many of the factor altogether. While this practice is satisfactory in the design of low-speed gears, it is not always safe design for high-speed gears and in gearing for machine tools.

GEAR BLANK. The design of the gear blank is a factor in both gear life and operating

noise level. Load distribution over the length of the tooth and the load distribution between teeth are influenced by the construction of the blank as well as by the accuracies with which the teeth are generated and the gears are mounted. The gear blank, as well as the teeth, deflects under service loads; it may warp from residual heat-treating strains; and it may allow the tooth profile to be displaced during the cutting process under cutter loads. The conditions producing the last type of distortion can be eliminated by proper care during the cutting operation. The responsibility for care during cutting should be shared by the designer and the machine operator because the operator may not visualize all of the adverse effects of the cutting loads.

The gear blank can be cast or forged. The process selected affects the range of blank designs that can be considered. The cast blank offers a great range of possibilities although it has the disadvantage of being subject to the types of sub-surface defects common to cast materials. Most sub-surface defects can be eliminated by good design and foundry practice. They can be detected by radiographic inspection. Forged gear blanks have the advantage of being relatively free of hidden defects although the forging process limits flexibility of design. In some cases these limitations will dictate the use of large, apparently unnecessary, quantities of metal to gain the desired rigidity.

Housing Cleanliness. There are several manufacturing details relating to the gear housing that can directly or indirectly influence the life of the gears. One of the most important is housing cleanliness. The gears and gear housing must be thoroughly cleaned by grit blasting before they are machined. The cleaning operation must remove all scale and sand from the exposed surfaces of these parts. If this foreign material is not removed, it may become dislodged in service and by abrasion destroy the gears and bearings.

Painting the housing surfaces without cleaning does not give adequate protection against the dislodgment of foreign matter. The paint fails and dirt enters the lubricant. Unpainted housings are likely to rust. During the operation of the speed reducer, this rust becomes dislodged from the surface of the housing and enters the lubricant. As rust is an abrasive, it causes premature bearing failures. Rusting is caused by the moisture in the air condensing on the walls of the gear housing.

The severity of rusting is dependent on the design of the housing venting system. The most pronounced rusting condition will be found in housings, which are made without vents, and the least adverse rusting conditions will be found when the housing is fitted with large vents located in the highest section of the gear housing.

Shaft Deflections. Lack of rigidity in the shafting carrying gears can cause unequal load distribution across the gear face. A shaft which is designed to operate within a safe stress range can deflect sufficiently to cause dangerous concentrations of tooth load and thereby cause the gear to fail through tooth wear or tooth breakage. This condition is most likely to occur when heat-treated alloy steel shafting is used because the allowable working stresses are higher so that for a given section a greater deflection results from the increased load.

In analyzing shaft deflection in gearing, it must be remembered that these deflections fall in three distinct classes and have three different effects. The first class is the linear deflection, the second class is the slope deflection, and the third class is the torsional deflection. If a gear is located in the center of the shaft span and is subjected neither to an axial load nor to external deflection loads, it is apparent that its movement would be linear. This movement would cause an increase in the center distance, but would not disturb the tooth load distribution materially.

If the gear location on the shaft is such that the shaft deflection tends to rotate the center line of the gear from its static position, slope deflection prevails. This type of deflection tends to concentrate the tooth load on one side of the gear face, thereby reducing the life of the gear. The effects of torsional deflection are most noticeable in pinions. If the pinion is weak in torsion, the tooth load will be concentrated on the power input side of the pinion. This condition will reduce the life of the gear train.

Deflections can be controlled by shaft size, shaft span between bearings, gear locations on the shafting, and helix angles. For

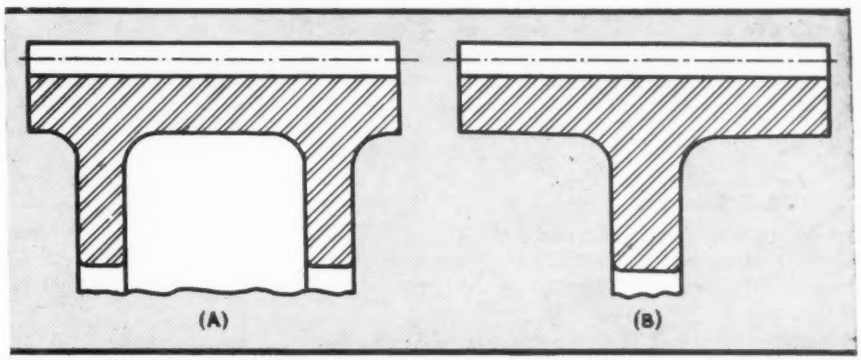


Fig. 2—(A) The H-arm construction has maximum resistance to distortion and blank deflection per pound of material. It is not suitable for forging. (B) The center-web type is used for both cast and forged gear blanks.

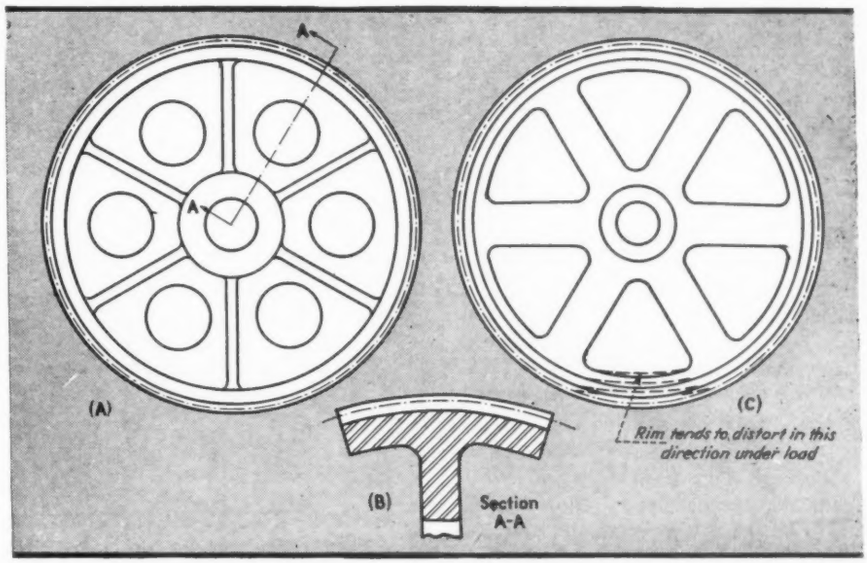


Fig. 3—(A) Resistance to deflection by the center-web type can be increased either by a thicker web or by ribs as shown. (B) Rim of center-web type, which has tendency to distort in the manner indicated under heat-treatment, can be prevented by greater rim thickness and by choice of material of suitable hardness. (C) Where rim section is small compared to span between spokes, rim may distort as shown by dotted lines or even break through under heavy loads.

herringbone and spur gearing, all of these factors, with the exception of helix angles, can be used to control deflection.

Deflections can be avoided to some extent in single helical gearing by choice of the hand of the helix angle in relation to the direction of rotation. A typical example of controlling deflections by controlling the relationship between the hand of the helix and the direction of rotation is shown in Fig. 5. For rotation in direction A the deflections in the intermediate shaft that result from the forces produced by the large gear add to the deflections produced by the axial components of the pitch line load on the two opposed single helical gears. When the gear train is rotated in the opposite direction, these deflection forces are opposed and the resultant deflection is reduced.

Another common type of failure related to shaft design is shown in Fig. 6. This shaft failed at a shoulder where the sharp corner caused an increase in the stress concentrations. The chemical composition of the material used in the gears and pinions can have a great effect on the service rendered by the gear train. Also, the heat-treating practices used in hardening the material will have an effect on the performance of the gear train. Surface hardness as measured by any one of the conventional hardness testing instruments will not give a true indication of the desirability of the material for use in gearing for, surface hardness is a measure neither of the degree of freedom from residual heat treating strains nor of the impact strength of the material.

The same surface hardness can be produced in a small shaft of plain carbon steel that can be obtained with a medium carbon alloy steel. Both steels can be heated to a temperature above the upper critical temperature, oil quenched, heated to some other temperature below the critical temperature range and air cooled. In the alloy steel shaft, the temperature to which the steel is heated after quenching to give the same hardness as that obtained in the carbon steel shaft will be much greater. As a result of the higher temperature the residual stresses remaining in the alloy steel shaft will be less than those in the carbon steel shaft. The degree of difference will depend on the size of the part and the property of the alloy steel to harden. The advantage of the alloy steel increases as the size of the shaft increases. Since warpage of a gear blank can cause a departure from ideal tooth contact, alloy steels can offer some relief from this condition when it is necessary to use constructions that are subject to distortion after cutting.

IMPACT STRENGTH. Basically, alloy steels have another advantage—higher impact strength. The usual methods for measuring impact strength use specimens which contain notches. These specimens are ruptured with a falling weight and the impact is measured by recording the amount of energy absorbed in rupturing the specimen. Since a notched specimen is used, this test is in reality a measure of the resistance of the material to crack propagation. As fatigue failures always start at a crack or a cleavage plane, the impact strength of a material is in reality a measure of the practical resistance of the material to fatigue failure.

In a paper by G. O. Riegel and F. F. Vaughn, *Practical Application of the Notched-Bar Impact Test*, A.S.T.M. Symposium On Impact Testing, June 1938, service failures in tractor gearing were correlated with impact strength. The data given in the paper indicate that when impact strength is low gear failures could be expected. Other impact tests have been made of alloy steel pinions that failed in service to determine if a correlation could be drawn between these service failures and the impact strength of the material. In each test the impact strength of the material was found to be low. It was also found that the lack of impact strength was caused by defective heat treatment. One of the pinions investigated is shown in Fig. 7. The physical properties shown in Table I are typical of all the pinions examined.

In Table I it will be noted that in each steel the impact strength was improved by another heat-treatment in the laboratory. The reason for this improvement was that the manufacturer's heat-treating procedure was faulty. The manufacturer's holding time at temperature during the quenching

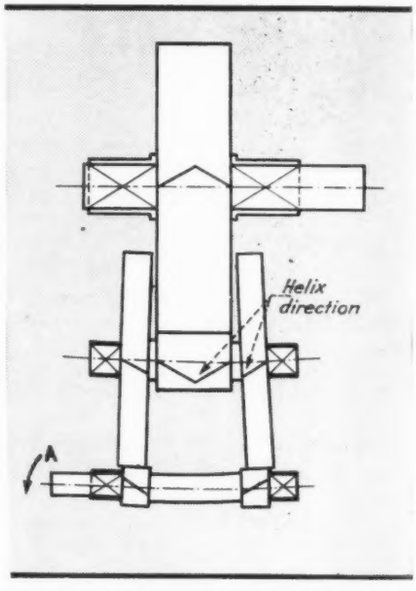


Fig. 5—The hand of the helix can be chosen with regard to the direction of rotation so as to reduce the unbalanced component of the forces.

Table I—Physical Properties of Alloy Steel Pinions Before and After a Second Heat-Treatment in the Laboratory

Heat-Treater	Tensile Strength, lb. per sq. in.	Yield Point, lb. per sq. in.	Elongation, percent	Reduction in Area, percent	Izod Impact, ft.-lb.	Brinell Hardness No.
Manufacturer	151,000	134,000	15.5	53.9	24	321
Laboratory	161,000	142,500	16.0	55.5	58	321
Manufacturer	144,000	128,000	13.5	49.7	17	293
Laboratory	145,000	145,000	17.0	56.8	58	293

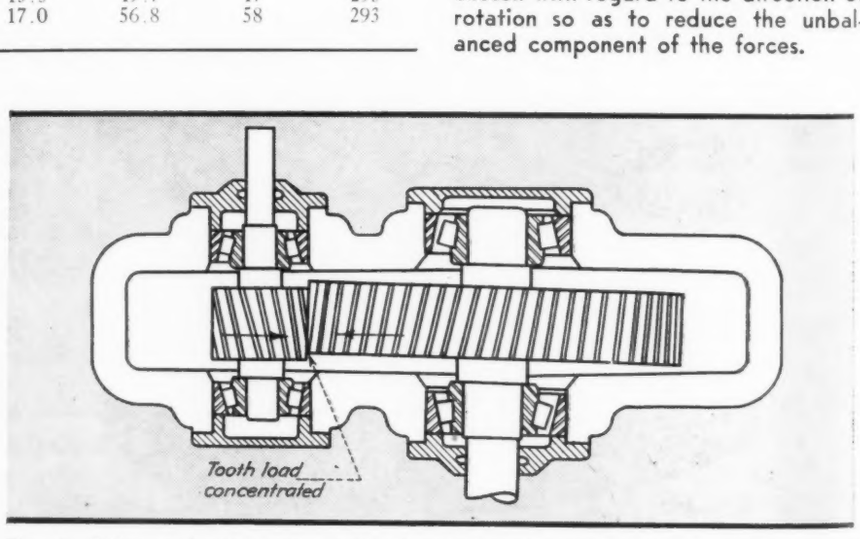


Fig. 4—Ribs can be incorporated in the design of gear housing so as to prevent housing distortions from axial thrust of single helical gears.

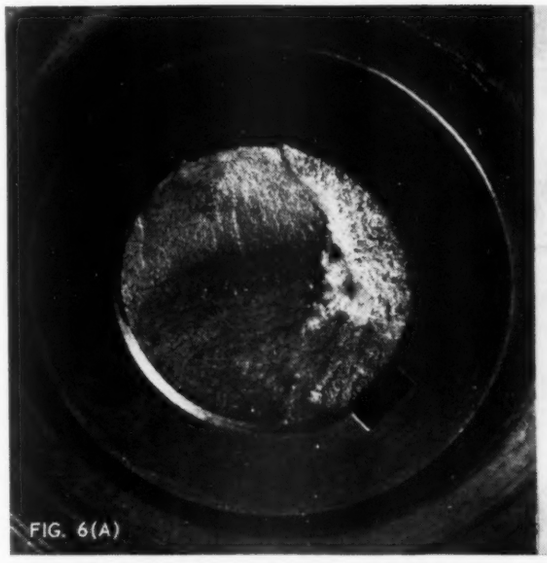


FIG. 6(A)

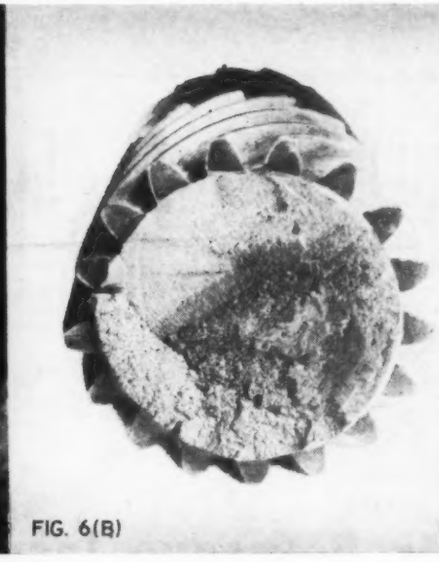


FIG. 6(B)

FIG. 7

Fig. 6—Two views of a shaft that failed through fatigue. A large, smooth radius at the shoulder will help avoid this type of failure. Fig. 7—Type of pinion failure that occurs when metal of low impact strength is used.

(first) heating was too short to place all of the constituents of the steel in solution. The same condition can be caused by ineffective quench, or by any one of several steel defects such as dirt in the steel. If the steel is defective, the low impact strength of the material cannot be corrected by heat treatment.

Length of Tooth Contact. One refinement common to high speed gearing that can be advantageously applied to low speed gearing is control of the length of the line of tooth contact. The amount of tooth contact is a function of the pitch diameters of the gear and pinion, face width, pressure angle, helix angle, outside diameters of the gear and pinion and the circular pitch. Some of these quantities can be varied by the gear designer to get optimum proportions. The results of an analysis of a large number of gear trains definitely indicate that minimum variation in length of tooth contact insures quieter operation and longer gear life.

In many gear trains the durability of all gears can be increased by using the long and short addendum type of involute gear teeth. The usual procedure in applying this type of tooth form to speed reducing gear sets is to enlarge the outside diameter of the pinion and decrease the outside diameter of the gear by the same amount. The teeth are cut to standard depth. The use of this system has the advantage of reducing pinion tooth undercut, discarding the sensitive portion of the involute and advantageously adjusting the lengths of the arcs of approach and recess. As the amount of undercut is decreased or eliminated, the base thickness of the tooth is increased and the tooth strength is increased.

The most difficult portion of the involute curve to generate accurately is that portion near, and at, the base circle. Also, specific sliding is high along this portion of the involute, and reaches infinity at the base circle. When the outside diameter of the pinion is increased and standard tooth depth is maintained, the tooth profile is removed from the base circle by an amount equal to half the increase in outside diameter. The result of this shift along the involute curve is that some of the more sensitive portion of the involute is discarded. Enlarging the pinion and decreasing the gear a proportional amount has practically no effect on the overall length of tooth contact. It does shorten the length of the arc of approach and increase the length of the arc of recess. This decreases the tendency of the tips of the driven teeth to dig into the flank of the driver teeth under heavy loading and destroy the oil film in front of the teeth. Also it increases the tendency for the tips of the teeth to carry oil under the teeth during the arc of recess. The principal advantage is increased tooth strength and this increase in tooth strength can be the difference between success and failure of the gears.

Workmanship and Assembly. Perfectly designed and cut gears can be destroyed by poor manufacturing processes in the assembly and final finishing operations. It is customary to hold the backlash in a gear train that is subjected to a reversing or variable torque load to a minimum. In the manufacture of this type of gearing, it is common practice to cut the gear and pinion a thousandth or two oversize and lap the gear and pinion together to obtain the desired amount of backlash. When properly carried out, this practice is satisfactory. If the amount of stock left on the gear is excessive, the lapping process will destroy the tooth profile. This condition is influenced by two factors; namely, the nature of the cutting action of the lapping compound and the fact that it is physically impossible to maintain a uniform distribution of the lapping compound across the gear face.

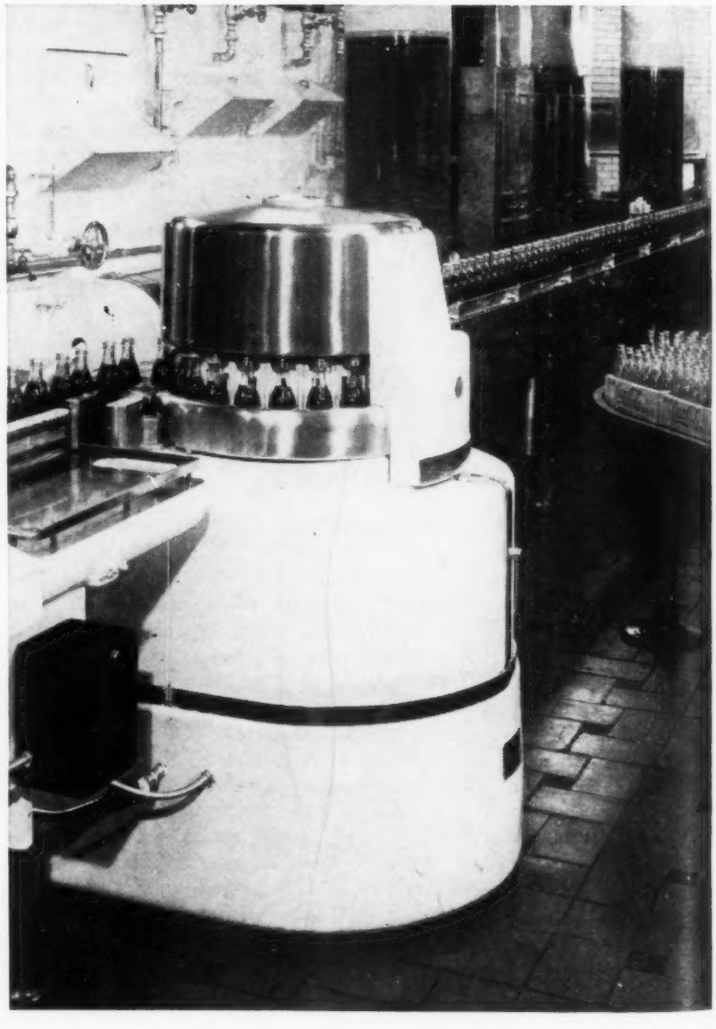
As the tooth action at the point of contact varies with the relative distance from the pitch line, and a certain amount of sliding contact exists at all points along the tooth profile except at the pitch line, the relative amount of material removed by the lapping process varies over the tooth profile. Within certain limitations this is an advantage, because it decreases the possibility of any point on the tooth surface projecting beyond the involute curve and affecting the action of the motion established by the helix tract. If the amount of material removed is excessive, the result will be the destruction of the proper involute action of the gear teeth.

Another manufacturing detail that can affect the durability of the gears, is the procedure used in mounting the gears on their shaft. If the gear is pressed onto the shaft with a large amount of interference between the shaft and the hole, the gear blank may be distorted during this operation, because of the stresses developed by the interference fit. If the gear hub is short compared to the shaft diameter, the gear may be pressed on with a slight angle between its plane of rotation and the axis of the shaft. Also, if the gear is cut on a false shaft or arbor, the center of the gear during cutting may not coincide with the center of the gear after it is pressed onto its shaft. To prevent the occurrence of these errors, the gear should be pressed onto its shaft before cutting, and it should be revolved about the true center of the shaft during the cutting operation.

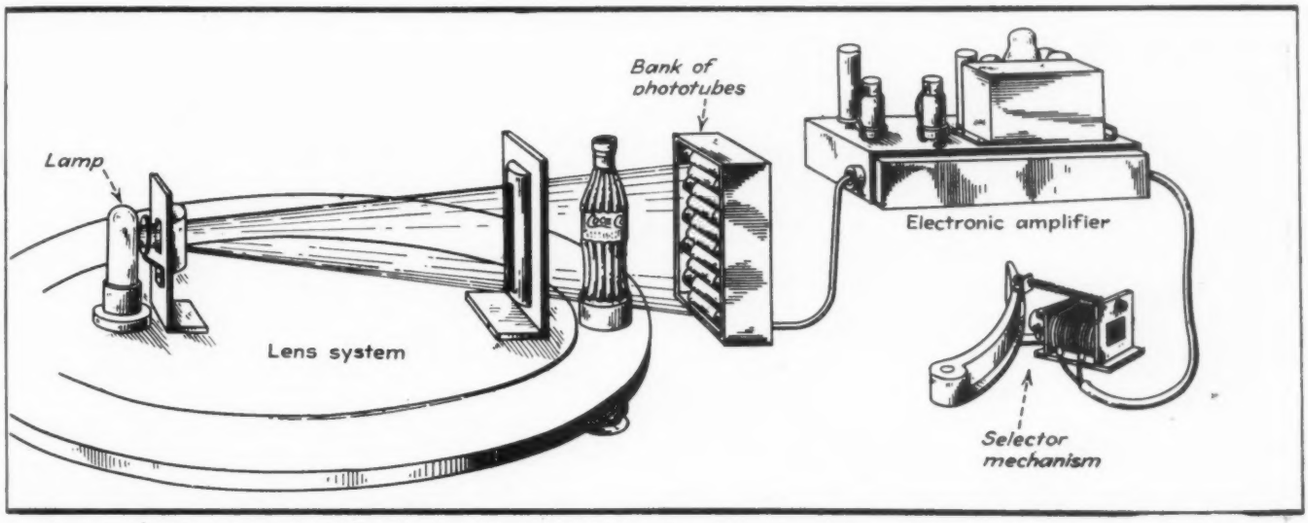
PRODUCT DESIGNS

Automatic Beverage Inspection Machine Electronically Guards Quality

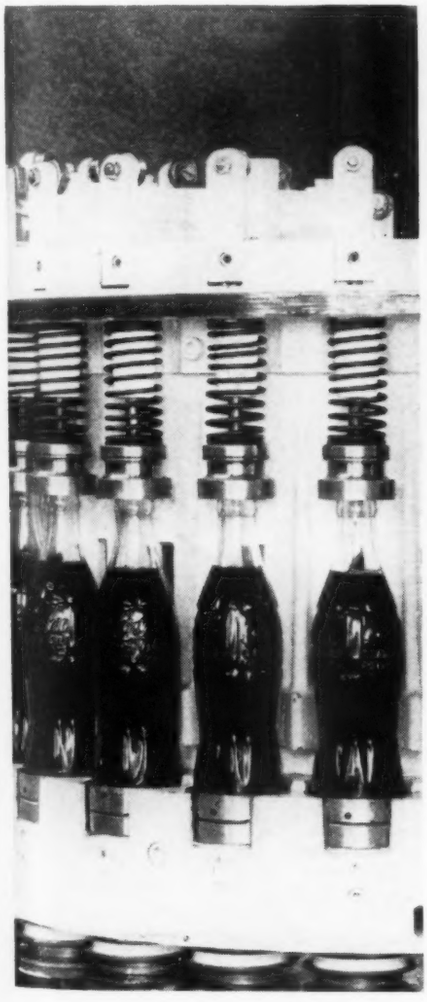
RESEARCH AND DEVELOPMENT on electronic inspection of bottled Coca-Cola was begun in 1934 by the Research Department of the Coca-Cola Company. Initially the development was under the direction of Dr. W. P. Heath and, later under B. G. Copping, chief engineer, with the Radio Corporation of America assisting particularly in the solution of the electronic and optical problems. In 1940, the Radio Corporation accepted responsibility for completing the development and the design of a commercial automatic inspection machine with the Coca-Cola Company continuing its active interest. Styling and appearance were under the direction of C. Louis Otto of Raymond Loewy Associates. In 1941, the Coca-Cola Company and Radio Corporation of America began extensive field tests in representative Coca-Cola bottling plants which have resulted in the announcement of the successful completion of the machine for commercial use.

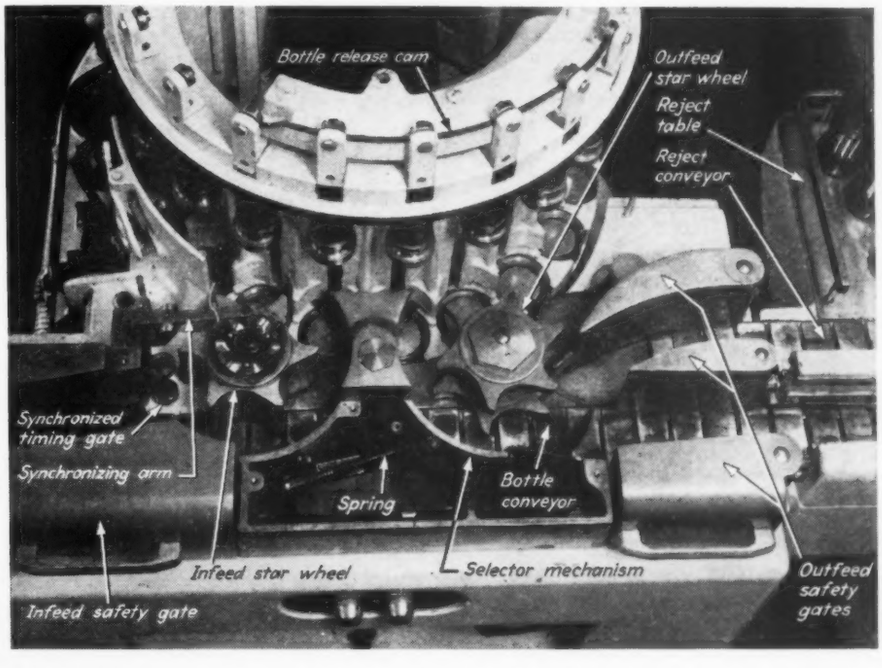


Automatic inspection of filled Coca-Cola bottles detects foreign matter in the contents of the bottle, even though it be of minute size or transparent in nature. Known as the RCA Automatic Beverage Inspection Machine for Coca-Cola, it combines and coordinates electronic, optical and mechanical units. Designed to be installed on any type of conveyor in use in Coca-Cola bottling plants, the machine will inspect up to 140 bottles per minute. Operation is fully automatic and requires only periodic attention by the operator. Failure of the inspection system and improper or insufficient rotation of any bottle will result in rejecting the bottle or in the stopping of the machine. Safety devices protect the machine from damage when jams occur in the conveyors and from broken bottles.



Inspection system consists of a lamp and lens that sends a strong narrow beam of light through the length of the bottle into a bank of phototubes. Sudden small changes in light on the phototubes, that are connected to an electronic amplifier, result in the operation of a selector mechanism that rejects the bottle. Prior to inspection, bottle is spun at high speed so that the contents spin with the bottle. Just prior to inspection, bottle is quickly braked to a complete stop and the contents continue to whirl rapidly during the inspection period. Particles moving with the liquid through the light beam are detected. Since the bottle is stationary with respect to the light beam during the inspection period, ribs, letters and imperfections in the glass are not causes of sudden changes in light and do not result in rejection.



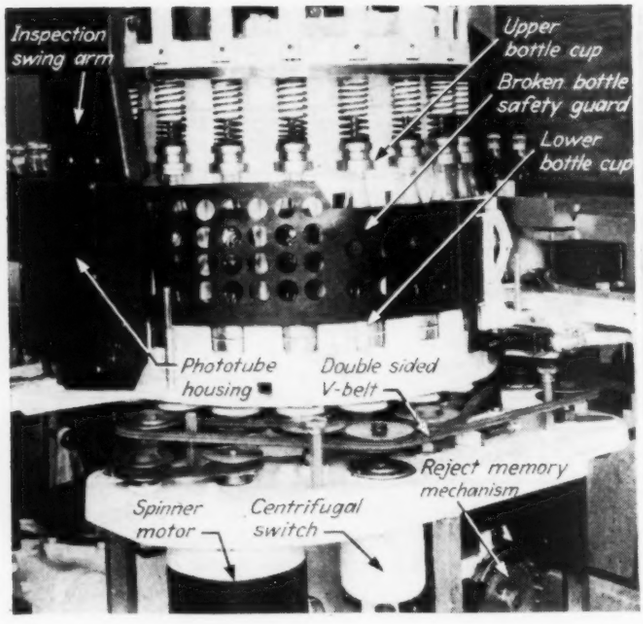


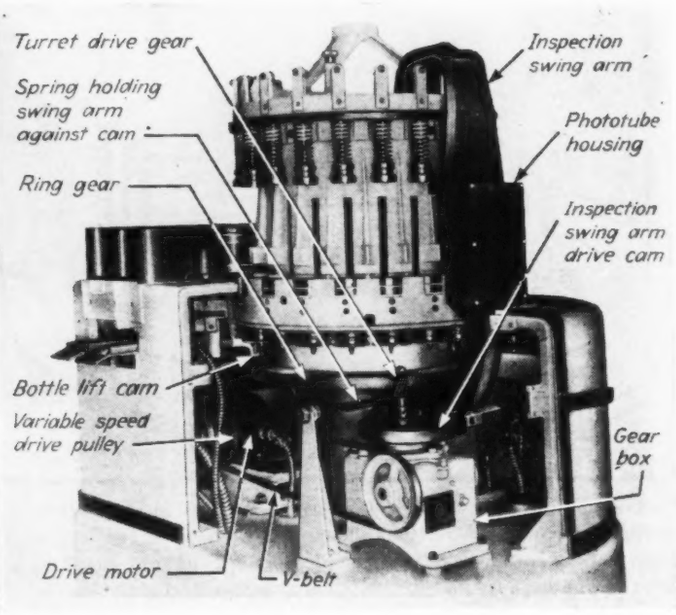
Synchronized timing gate controls entrance of bottles from the conveyor through the star wheel into the machine. Bottles are removed from the machine by a second star wheel. Two-position selector arm acts to deflect rejected bottles to the reject table. Safety gates guard against jamming at the infeed and outfeed conveyors.

Bottles are held between upper and lower cup assemblies. Upper cup assemblies are raised by a cam to permit bottles to enter and leave the turret. At the same time a bottle lift raises the bottles above the lips of the lower cup assemblies. Pulleys attached to the lower cup assemblies are driving connections during spinning. Bottom surfaces are used for braking to a stop, before inspection.

(*continued on next page*)

Automatic Beverage Inspection Machine (continued)

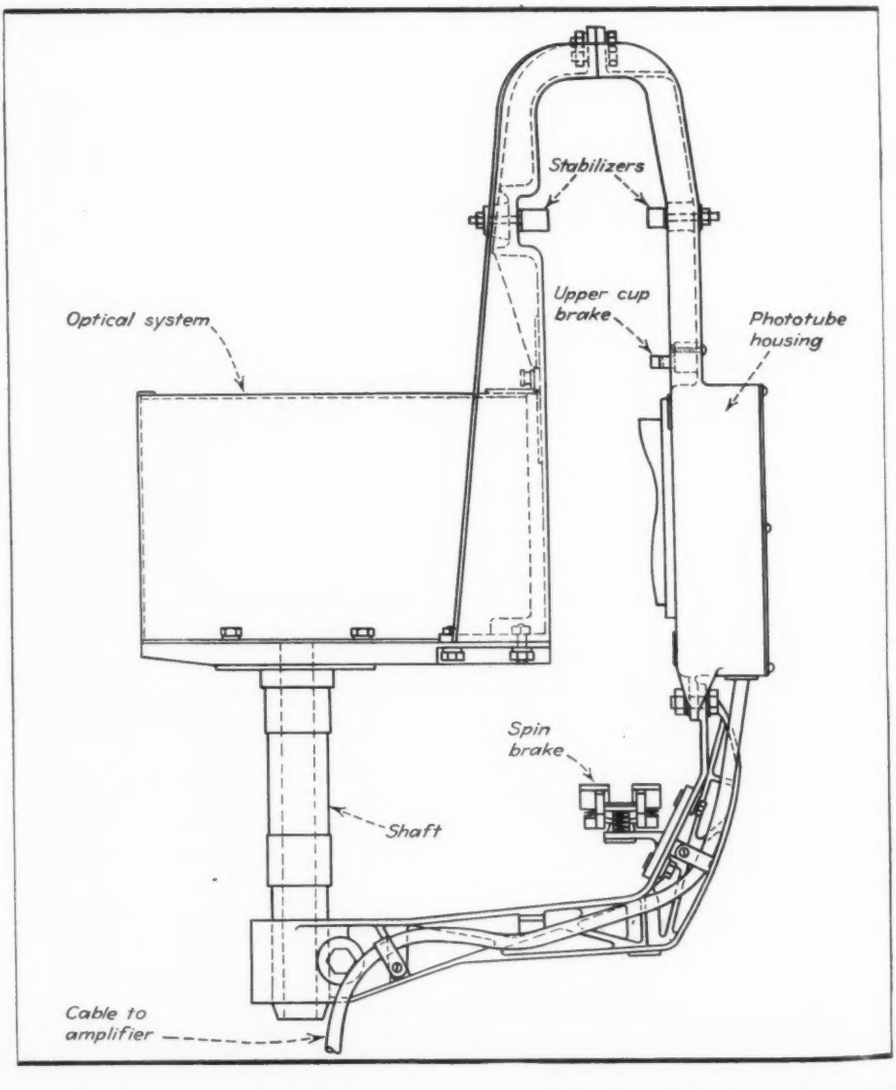


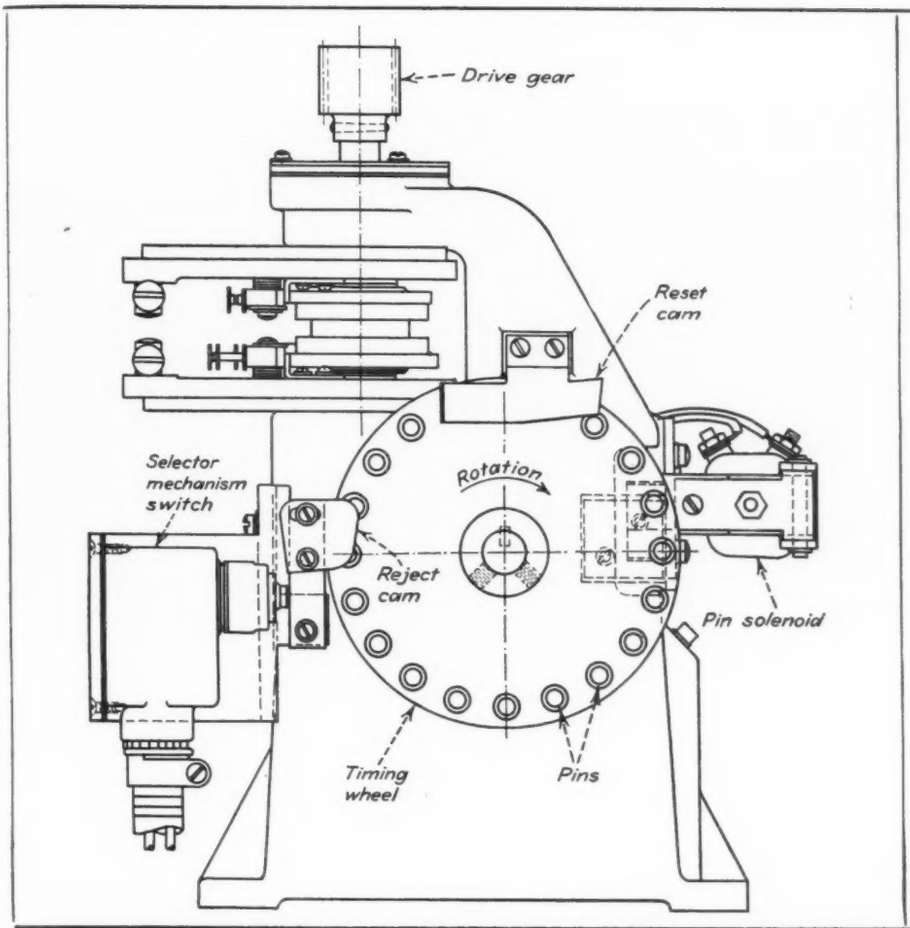


Double sided V-belt drives the bottles during spinning. Centrifugal switch will stop machine if spinning speed is not correct. Spinner motor is located below V-belt drive.

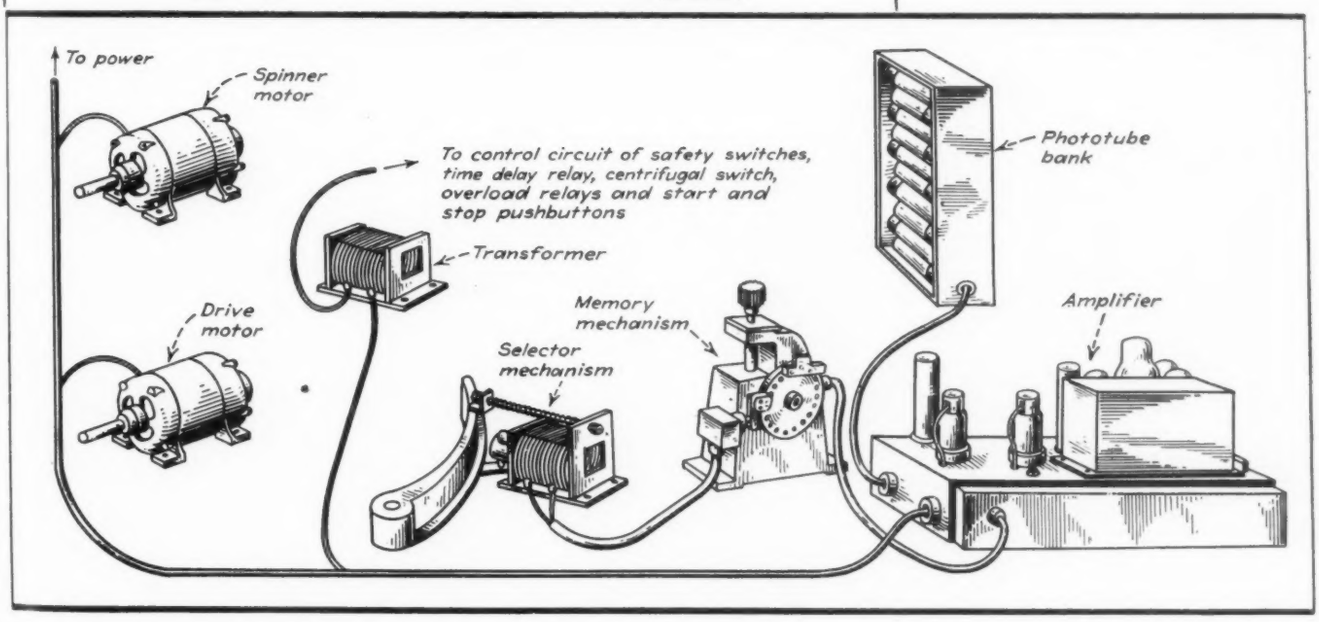
Motor drives turret and inspection swing arm cam (above right) through a variable speed V-belt pulley and a gear box. Cam imparts an oscillating motion to the inspection swing arm so that it can be interlocked with each turret position as that bottle is inspected.

Inspection swing arm is attached to the top and bottom of a shaft centrally located with respect to the turret. Optical system is mounted on a plate attached to the centrally located shaft. Inspection brake, upper cup brake and stabilizers are attached to the inspection arm and interlock the turret, bottle and inspection swing arm during the inspection period. Stop brake consists of two brake shoes equalized through a system of arms by three springs. Universal joint allows brake shoes to seat accurately on the braking surface of the spin drive pulley.





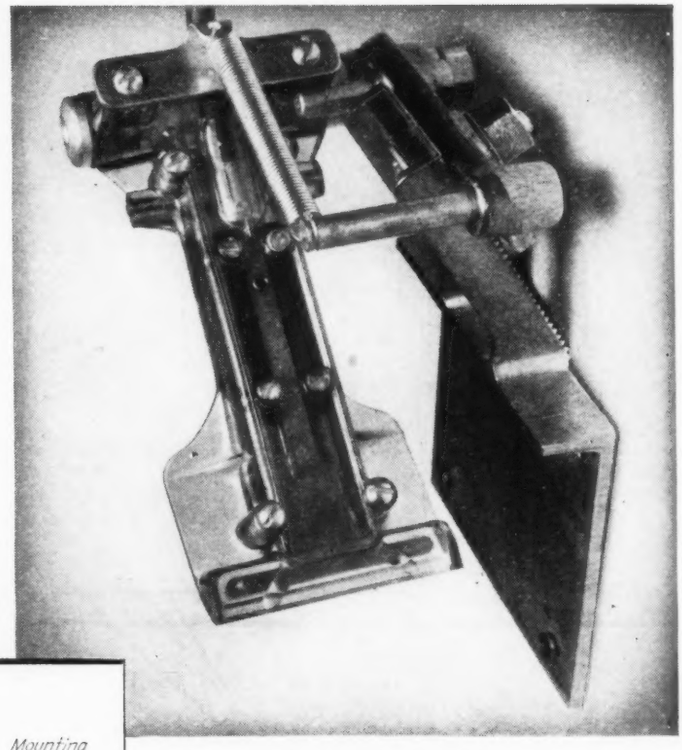
Memory mechanism is driven from turret ring gear. Timing wheel, having as many pins as there are turret stations, rotates at the same speed as the turret. Rejection of a bottle by the inspection system operates a solenoid that pushes out a pin in the timing wheel. When the pushed out pin strikes the reject cam it operates a switch, controlling the selector arm motion, exactly at the moment when the bottle is leaving the outfeed star wheel. Thus, the unsatisfactory bottles are guided to the reject table. Reset cam located at the top of the timing wheel returns pins to their orginal position.



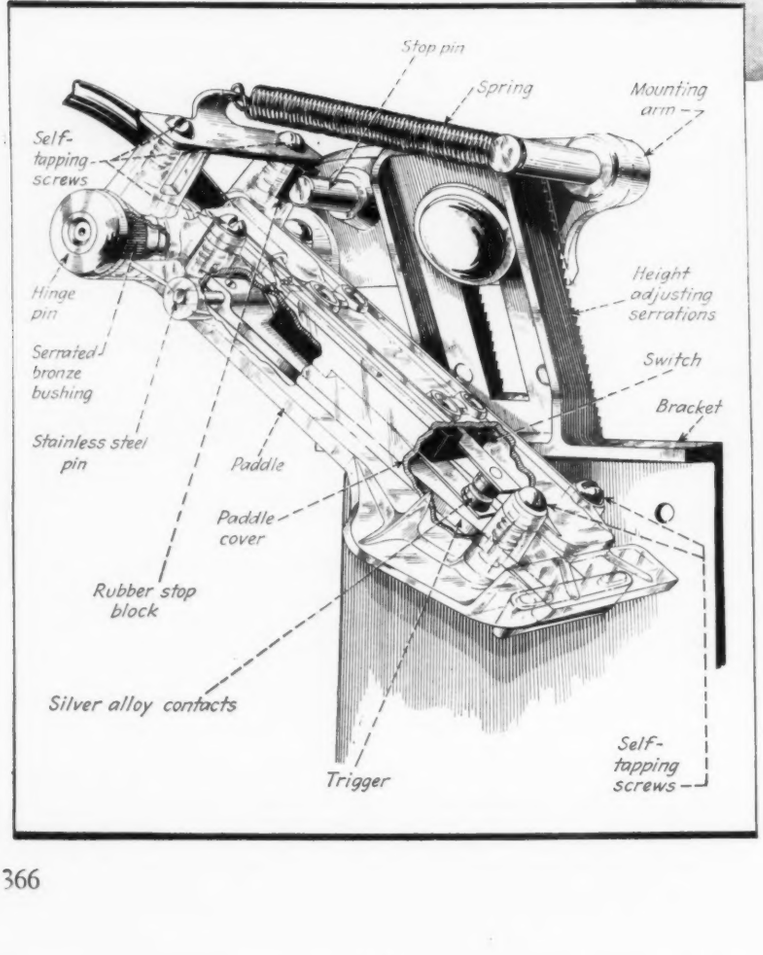
Control circuit includes safety, start, stop and centrifugal switches, the time delay and overload relays. The centrifugal switch and the time delay relay are open. Centrifugal switch closes when spinner V-belt comes up to speed. Time delay relay is actuated when RCA-6H6 tube in amplifier heats up, thus assuring amplifier is in operation before machine is run. Particle in bottle causes sudden changes in current of phototubes. This signal is amplified using RCA 1620 pentodes and triggers the RCA OA4-G thyratron. Current through the thyratron opens reject relay and releases solenoid that sets pin in the memory mechanism. Amplifier power is controlled by main line circuit breaker and remains connected even though the control circuit should be opened and stop the machine. On restarting, bottles, not properly spun before inspection, are rejected by placing a bias on the RCA 1620 pentodes thus actuating the RCA OA-4-G trigger tube for an instant after the machine is started. Inspection system is checked before each bottle is inspected.

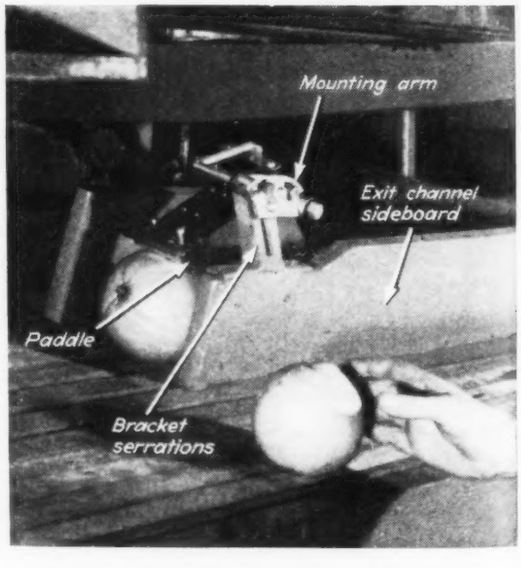
Electric Citrus Fruit Counter Design Uses Plastic and Die-Cast Parts

COUNTING FRUIT during processing and the obtaining of an accurate count as soon as the run is over is possible with the Food Machinery Corporation's electric fruit counter. Designed primarily for oranges and other citrus fruit the device should find use in other applications. Consisting of two parts, the counter and the recorder, this device gives an accurate count as soon as the run is over and no time is lost by the usual shutdown required to estimate the run. Fruit lifts a small paddle-like device and closes an electric circuit which operates a relay that advances an ordinary mechanical counter of the printing type.

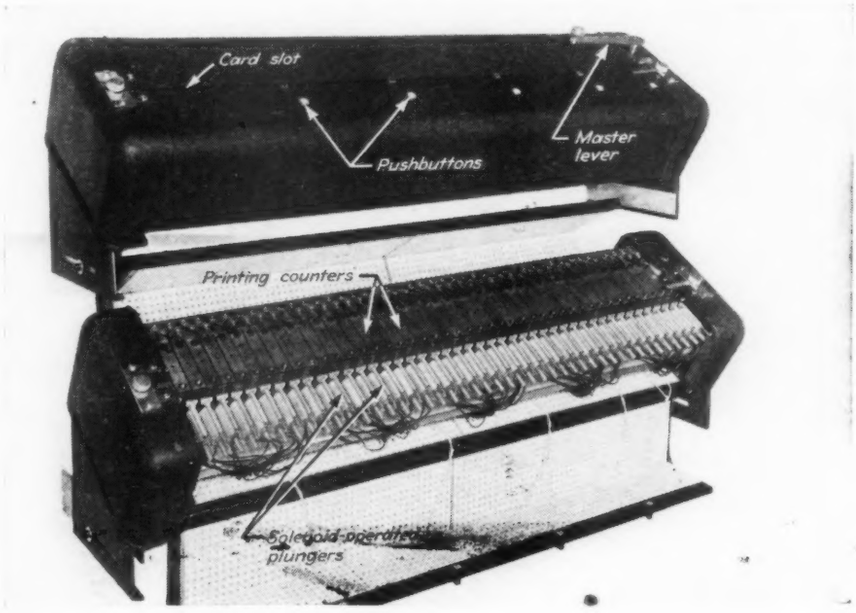


Contact mechanism uses transparent plastic and die-cast aluminum alloy parts. Paddle is of plastic having two serrated bronze bushings at the hinge pin. Counter switch is screwed to paddle top. Trigger is an aluminum alloy die casting pivoting in the paddle on a ground stainless steel pin and makes the electrical contact. Bracket and mounting arm are also aluminum alloy die castings.



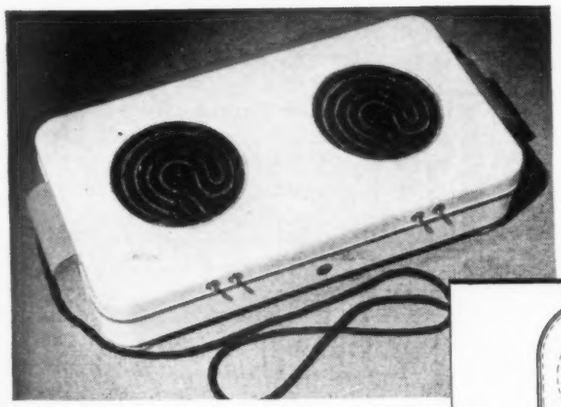


Bracket is screwed to the wooden sideboard of the sizing channel. Serrations on the bracket and mounting arm allow the paddle height to be adjusted to accommodate fruit of the size passing through the exit channel.



Counter consists of a bank of printing counters opposed by solenoid-operated plungers. Record card is inserted in slot that runs the length of the cabinet and is brought against horizontal and vertical stops. Pushbuttons energize the bank of solenoid-operated plungers at the card location. Inked tape runs full length of the cabinet and counters are set to zero reading by a master lever.

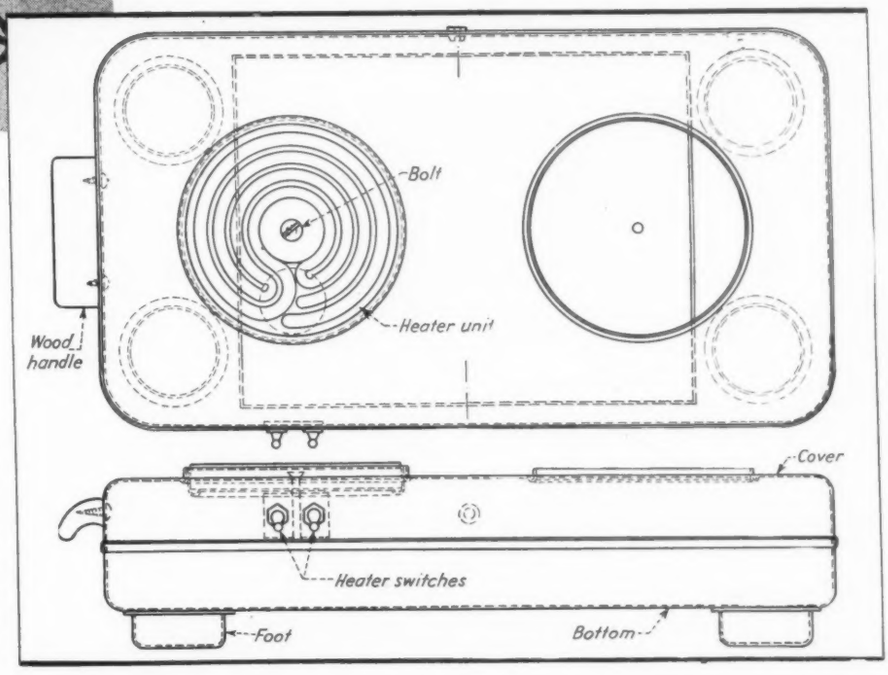
Electric Hot Plate Made from Steel Stampings



DESIGNED FOR APPEARANCE, strength, cleanliness and simplicity in operation is this new electric hot plate, designed by Benjamin L. Webster and manufactured by the General Aviation Equipment Company.

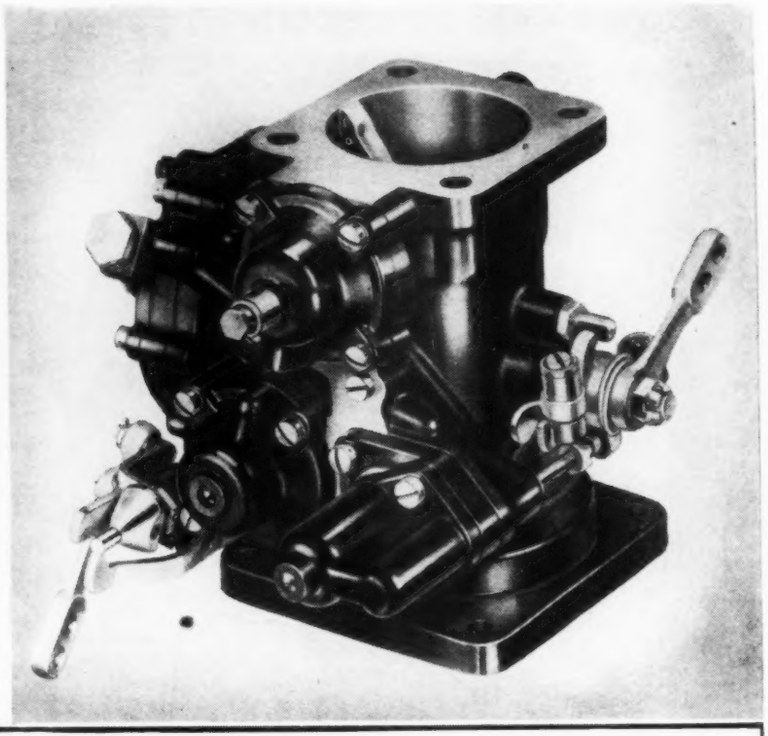
Each burner has two heats controlled by two switches. Design of the receptacles for the ceramic bricks is such that heat is not transmitted to the top of the unit. Finish is a heat-resistant, plastic lacquer.

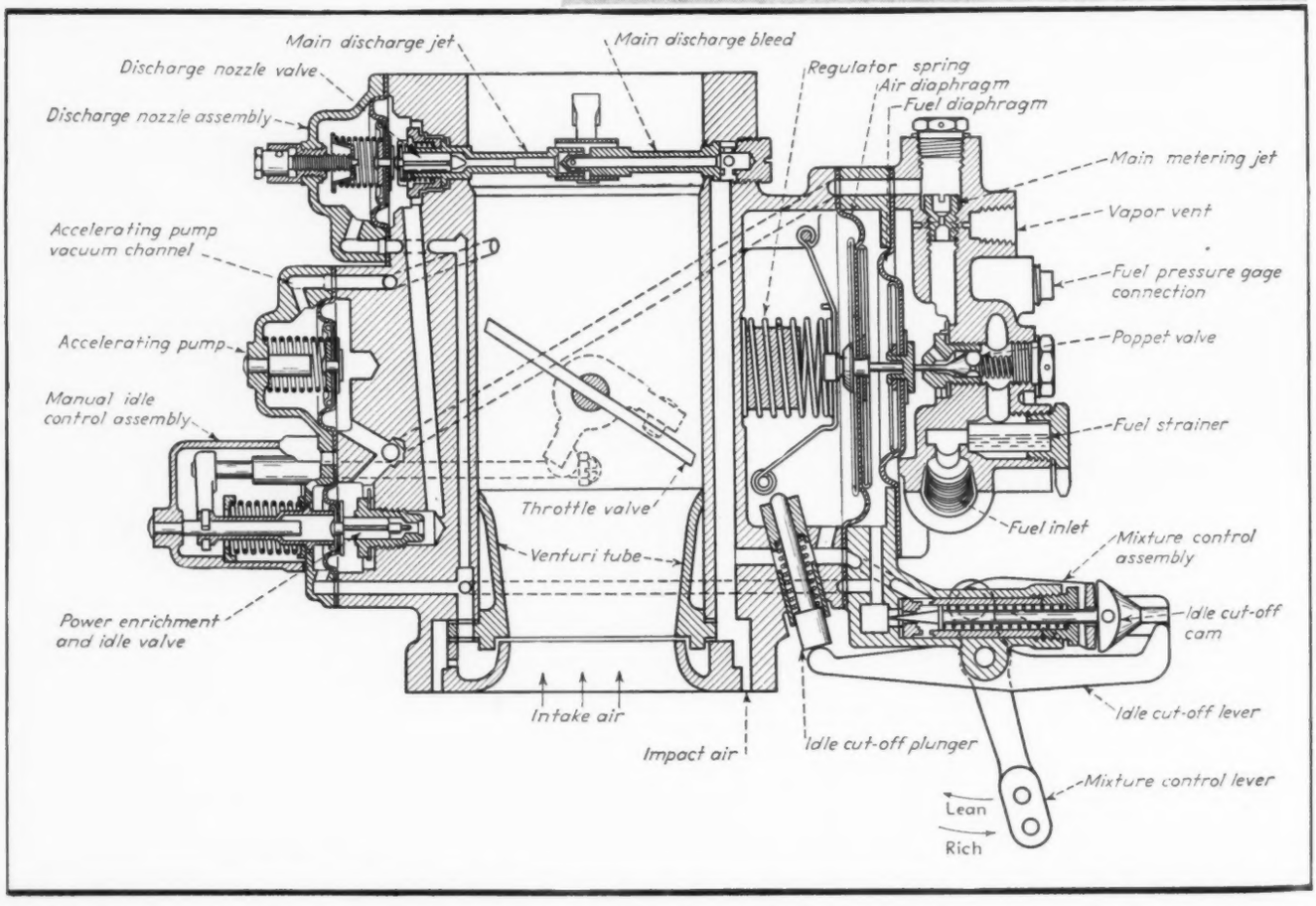
Steel stampings are used throughout the design. Top and bottom pieces are made in the same die. Feet are also steel stampings. Use of a single stamping for the top eliminates crevices found in many hot plates and leaves no places for food or dirt to collect. Wood handles allow the hot plate to be moved while operating.



Injection Carburetor For Light Aircraft

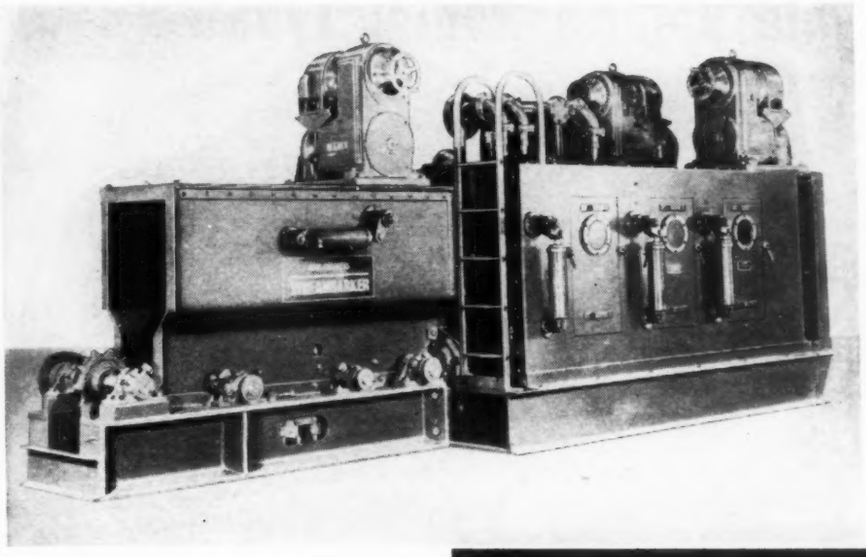
OPERATING ELEMENTS are compact, simplified and economical in design in the Stromberg PS Series Injection Carburetor developed for engines ranging from 50 to 500 hp. Produced by the Bendix Products Division, Bendix Aviation Corporation, these carburetors are distinctly different from previous carburetors of the same size and power range in that they use no vented float chamber or suction pickup from a discharge nozzle placed within the venturi tube. PS Series carburetors are of single barrel construction and can be mounted in any position.





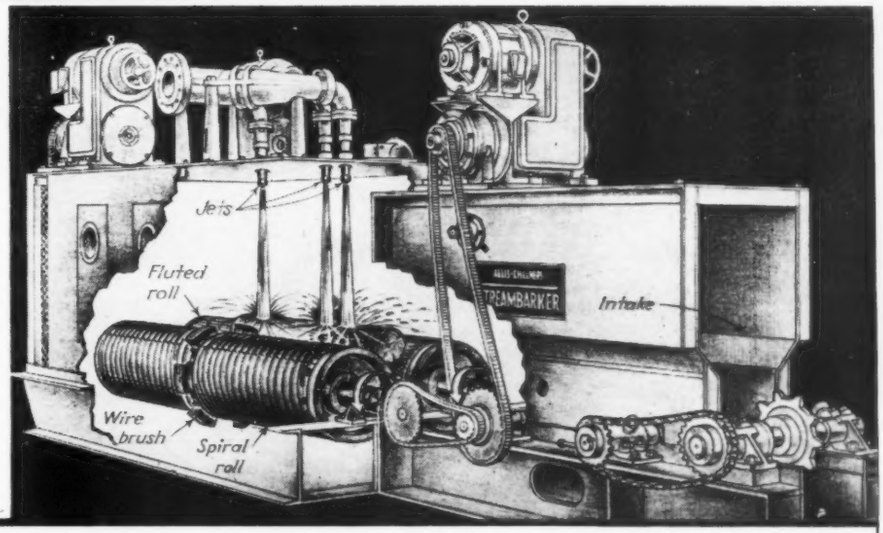
Fuel feed is through a closed pressurized system from the fuel pump to the discharge nozzle. Venturi serves only to create a pressure differential that is used to control the quantity of fuel. Accelerating pump is vacuum-operated by a single diaphram. Mixture control is manual and operates in combination with the idle cut-off. Fuel flow is automatically reduced to idling requirements when throttle is moved to idling position. Fuel lines and carburetor are always full even when the engine is stopped. Automatic mixture control and power enrichment are available but optional.

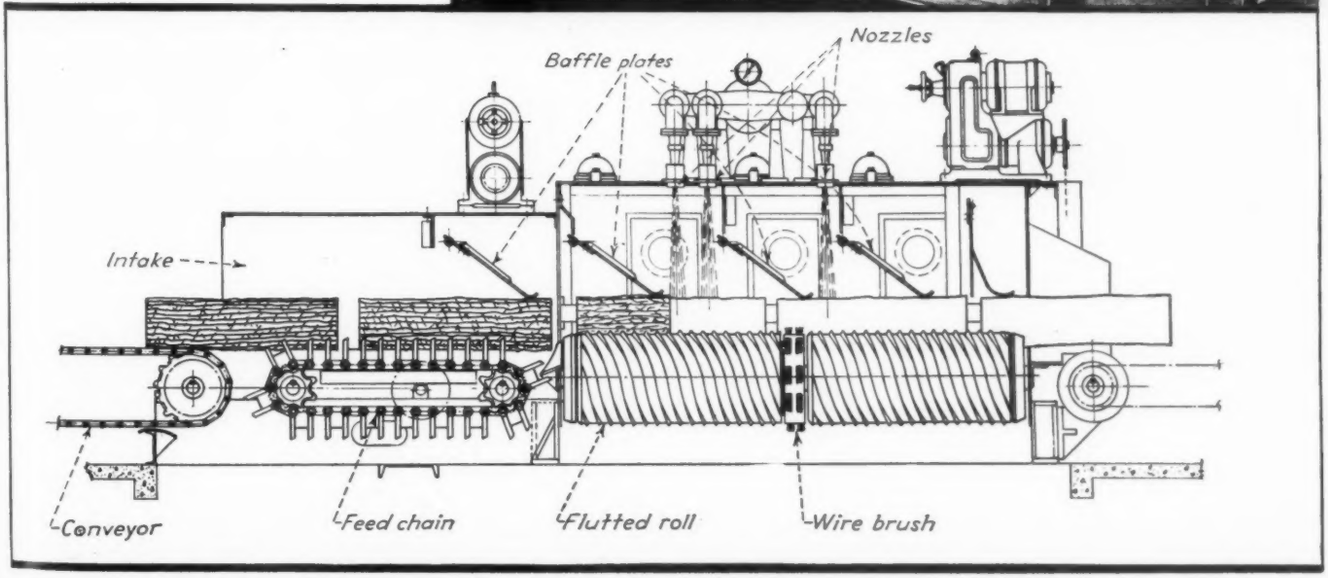
Machine Removes Bark From Logs



IN THE STREAMBARKER, developed by the Allis-Chalmers Manufacturing Company for standardized production, a 350 hp. electric motor drives a 4-stage centrifugal pump to furnish 600 gal. per min. of water at 650 lb. gage pressure to the water jets. Size of the Streambarker is 6 ft. 6 in. wide, 6 ft. high, 20 ft. long and weight 40,000 lb. Constructed of heavy welded steel plate and rigidly reinforced structural steel, the unit contains large watertight doors for easy accessibility to the inner working parts. Top-mounted waterproof floodlights illuminate the interior. Glass windows allow the entire action of the machine to be observed.

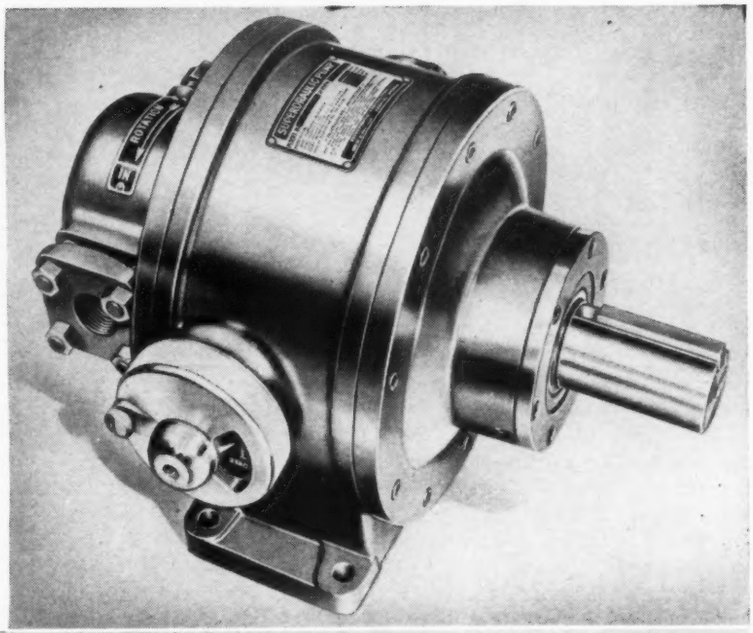
Logs are fed by a conveyor from the log pond or storage pile to the machine intake end. A chain conveyor then feeds the logs to the processing chamber. Two bed rolls, one fluted and the other spirally threaded, rotate in the same direction but at different speeds and convey the log under the high-pressure water streams. Wire brush centered on the spiral roll sheds the "redtail" bast fiber. Sharp threads on the rolls lay open the bark and assist the action of the water jets. Bark, dirt and waste water drop under the machine into a welded steel water hopper and sluice for disposal.



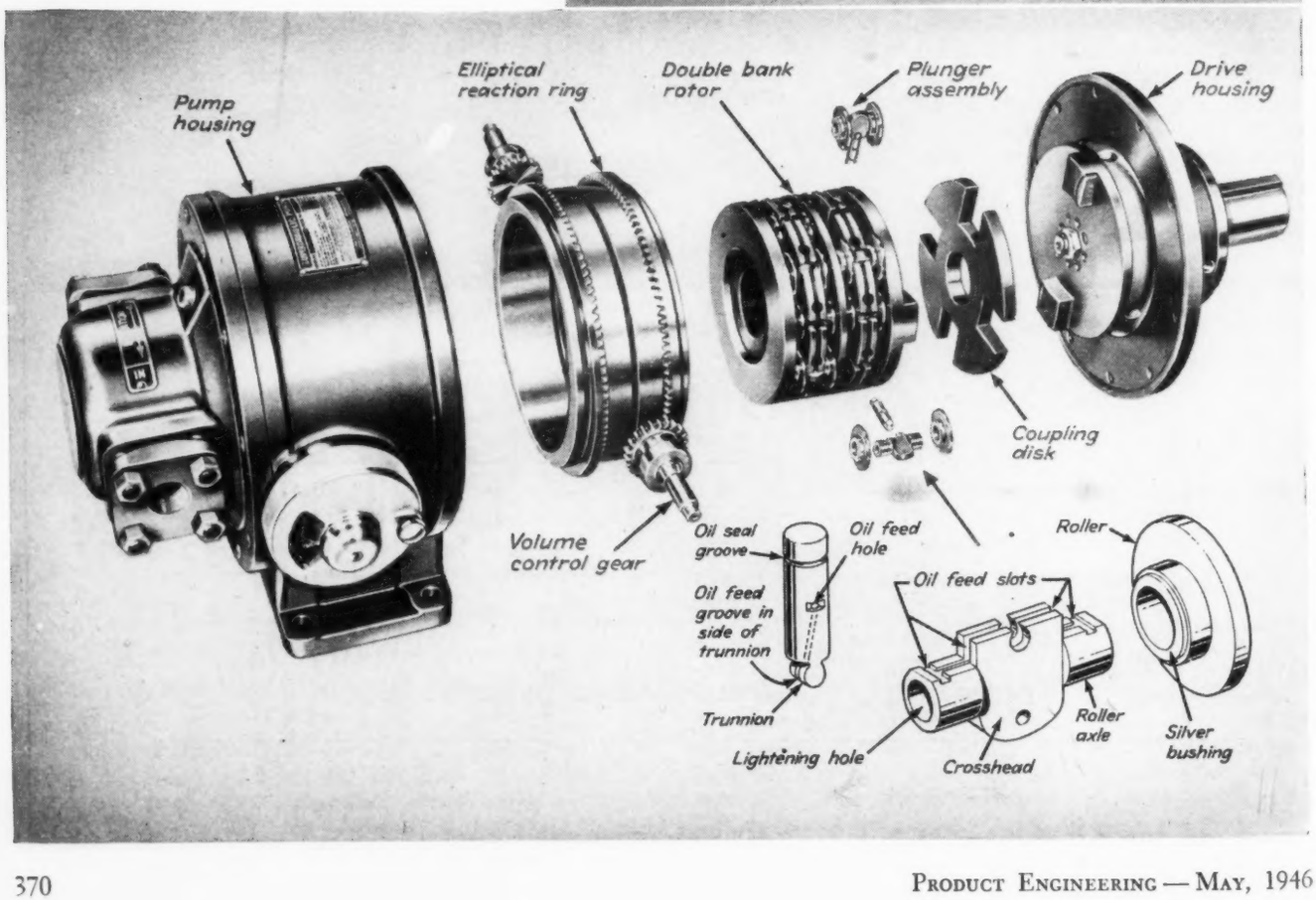


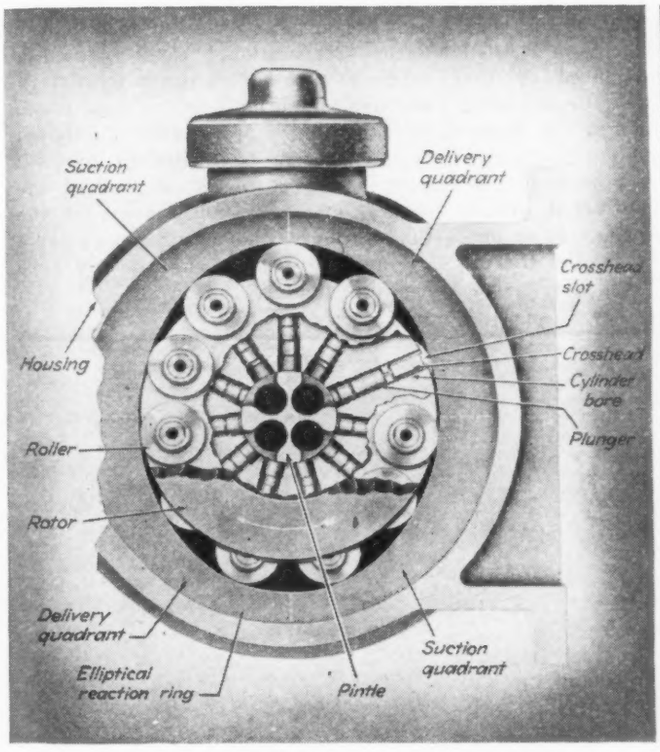
High-Pressure Pump of Compact Design Is of Variable Volume Radial Type

RADIAL TYPE PLUNGER PUMP of compact design has been developed for continuous duty 5,000 lb. per sq. in. operation. The Superdraulic pump is available in two models, a constant delivery model and a variable delivery model. Delivery at 1,200 r.p.m. is 0 to 17 gal. per min. Dial control, mounted on the side of the pump, indicates output volume.

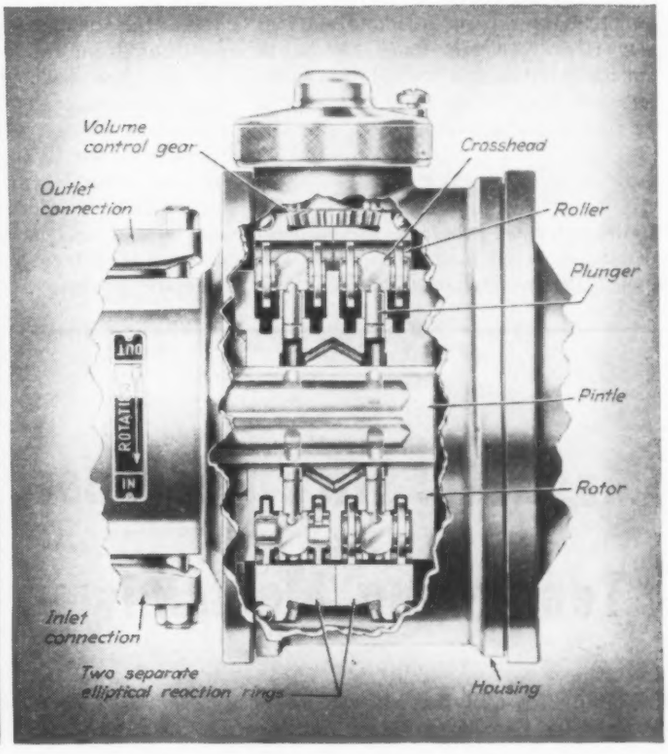


Double-bank rotor is driven by shaft through a coupling disk. Rotor has two banks of 11 plunger assemblies. Plunger is held in crosshead. Two rollers, one on each end of the crosshead, are held in contact with the elliptical raction rings by centrifugal force. Below speeds of 100 r.p.m. springs are used to hold plungers outward. Reaction rings are positioned for variable delivery by two volume-control pinions.





Plungers make two inlet and two delivery strokes per revolution (left). Trunnion mounting of piston in crosshead eliminates side thrust on the plunger. Rotor turns on a fixed pintle having ducts for both intake and exhaust.



Plungers in the two banks, connected by drilled passages, are arranged for parallel action (right). Volume control gear rotates the elliptical reaction rings, in opposite directions, to control the pump output.

Random Notes on Reconversion

Machine automatically seals an air-tight, moistureproof band of tape around lids of different sized cans, bottles and containers. Developed by the Package Machinery Company the machine has a speed of twenty-five containers per min. and an easy opening device for removing the tape. Designed primarily for coffee or baking powder containers, it will find use in packaging drugs, tobaccos and other items.

* * *

Die casting machine costing no more than a bandsaw or small engine lathe has been announced by the DCMT Sales Corporation. Machine uses prefabricated die sets that permit the manufacturer to make his own dies. Runs as small as 250 are said to be economical.

* * *

Purchase of the Anderson Stove Company, Inc. by Breeze Corporations, Inc. will accelerate the manufacture of the Anderson kitchen gas range. Sheet metal parts will be manufactured in Breeze plants in Newark and Elizabeth, N. J. and castings will be furnished by Foundry Service, Inc., also purchased by Breeze.

* * *

Rubber-covered potato basket to safeguard potatoes against bruising has been announced by United States Rubber Company. Basket, of open construction, is made of steel wire coated with rubber, weighs 3¾ lb. and holds 5/8 bushel.

* * *

Aluminum is used in the housing, hubs and brake shoes of a lightweight, heavy-duty truck axle. Developed by the Timken-Detroit Axle Company, the use of aluminum reduces the axle weight by 220 lb. when compared to the same axle with malleable iron parts. Cost of the aluminum axle is more than a conventional axle but reduction of unsprung weight should increase tire life.

* * *

Stainless steel, streamlined refrigerator car weighing 5,000 lb. less than conventional equipment will be placed in experimental service by Atchison, Topeka & Santa Fe Railway this year.

* * *

Hot meals prepared on Pan American Constellation-type clippers require two specially built electric whirlwind ovens to a galley. Ovens are capable of defrosting and finishing the cooking of twelve frozen blue-plate dinners, prepared by the Maxson Food Systems, every fifteen minutes.

* * *

Lightweight refrigeration unit will be installed in new Lockheed P-80, Shoot-

ing Star, airplanes. Unit is necessary to cool metal skin of airplane since it has a rise of 20 deg. F. at 350 m.p.h. from air friction and should the speed reach 650 m.p.h. a 60 deg. F. temperature rise would develop.

* * *

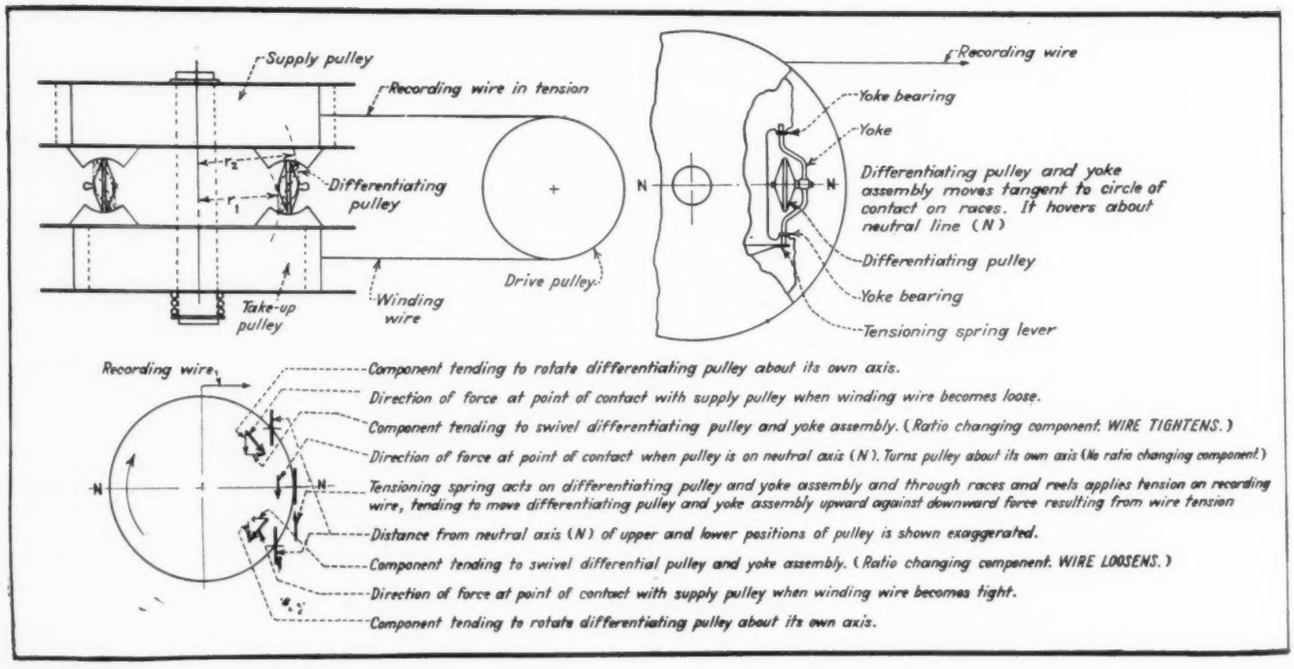
Aluminum-framed windows for residential use are being manufactured by the General Bronze Corporation.

Electronic tool, so sensitive that it will detect the unbalance in a rotating part resulting from the presence of a wisp of lint, has been announced by the Westinghouse Electric Corporation. Known as the Microbalancer the device was developed to speed the production of the variable time proximity fuse and the rotors used in gyroscopic instruments. The unit is incorporated in production machines built by the Gisholt Machine Company.

Entry into the photography field has been announced by the American Safety Razor Corporation. Camera has a magazine with sufficient film for sixteen pictures and is still in the final phase of laboratory preparation. Exposed magazines will be turned in to local dealers and processed at one of the several district establishments of the Corporation. Other new products are a nylon shaving brush and a new type cigarette lighter.

Magnetic Wire Recorder Tensioning Mechanism

PORTABLE MODEL magnetic wire recording machine, for home and office use, has been developed by the Brush Development Company. Size is approximately 14 in. by 11 in. by 8 in. deep and weight is less than 30 lb. Device consists of three essential elements; the recording head, the recording wire and the tensioning mechanism. Recording wire is a ductile wire electro-plated with a magnetic material. Tensioning mechanism is an ingenious differential device developed by A. L. Williams, president of the company.


Supply pulley
Recording wire in tension
Differentiating pulley
Take-up pulley
Winding wire
Drive pulley
Recording wire
Yoke bearing
Yoke
Differentiating pulley and yoke assembly moves tangent to circle of contact on races. It hovers about neutral line (N)
Differentiating pulley
Yoke bearing
Tensioning spring lever
Recording wire
Component tending to rotate differentiating pulley about its own axis.
Direction of force at point of contact with supply pulley when winding wire becomes loose.
Component tending to swivel differentiating pulley and yoke assembly. (Ratio changing component. WIRE TIGHTENS.)
Direction of force at point of contact when pulley is on neutral axis (N). Turns pulley about its own axis (No ratio changing component.)
Tensioning spring acts on differentiating pulley and yoke assembly and through races and reels applies tension on recording wire, tending to move differentiating pulley and yoke assembly upward against downward force resulting from wire tension
Distance from neutral axis (N) of upper and lower positions of pulley is shown exaggerated.
Component tending to swivel differential pulley and yoke assembly. (Ratio changing component. WIRE LOOSENS.)
Direction of force at point of contact with supply pulley when winding wire becomes tight.
Component tending to rotate differentiating pulley about its own axis.

Differential drive unit maintains constant wire tension and speed. Wire comes off the supply reel and is wound twice around the power pulley, driven by a small motor. Wire then goes through the recording head, then to the take-up reel. As the drive pulley draws wire from the supply reel a differential drive unit, connecting the supply and take-up reels, turns the take-up reel in a direction opposite to that of the supply reel thus maintaining constant torque on the take-up reel. A level-winding shaft insures an even spread of wire on the reels and prevents pile-up of the wire.

Design Calculations Are Not By Rule and Rote

IN TEACHING mechanical engineering design in our schools and universities, formulas are all too often presented as being physically exact relationships. Twelve of sixteen college machine design textbooks investigated state that unit tensile stress is equal to total load divided by cross-sectional area. The other four books come closer to the truth. They state that the average unit tensile stress is equal to total load divided by cross-sectional area, but neglect to make clear that it is the peak stress that determines whether or not the part will fail. None of the books explains the many conditions that cause the stress distribution over the cross-section of a part to be other than uniform. Nor is the student given any idea of the magnitude of variation in stresses across a given section.

Modern machine design textbooks, it seems, still cling to concepts that are valid in the design of civil engineering structures, but that are seldom applicable in designing moving parts in machinery. Mechanical engineering design textbooks place great emphasis on yield strength and factor of safety, give too little attention to fatigue failure, and fail entirely to point out the many influences that might cause an apparently well designed part to fail by fatigue.

None of the textbooks investigated properly explained how cold rolling, cold drawing, surface peening, carburizing and nitriding develop compressive stresses on the surface, thereby causing a stress pattern that may greatly strengthen the part against fatigue failure or cause it to break at a low load. Nor is it explained how machining, grinding and other operations create surface tensile stresses that greatly weaken the part.

Wrong ideas and concepts firmly established in a student's mind are extremely difficult to eradicate even by bitter experience. Chief design engineers frequently comment that young mechanical engineering graduates seem shocked and dazed when they discover that the classical equations, which they so laboriously learned, have extremely limited application except as help in visualizing ideal stress conditions. Then slowly it dawns upon them that formulas must be modified extensively and on an intelligent basis, that relatively few stress problems can be solved with any degree of exactness, and that endurance tests on the finished part are the only sure basis on which to determine the durability of the design.

Many, if not most, of the professors of machine design recognize that engineering schools and colleges face the task of renovating their design courses to improve the value of the students to industry. They will do well to consult freely with outstanding designers in industry in the building of machine design courses that emphasize the whys and wherefores of realities and not the rule and rote that does not conform to facts. And it should always be emphasized that even the rational equations are true only for non-existing ideal conditions and give results that are crude approximations of the true facts.

G. F. Nordenholt.

Materials and Manufacturing Processes Determine Economical Tolerances

A. L. KLEIN
Design Consultant, Douglas Aircraft., Inc.

A compilation of reasons why unnecessarily small tolerances should not be specified on engineering drawings. This material is presented from the economic, manufacturing, and assembly standpoint and furnishes the design engineer with concrete examples of what and what not to avoid.

TO ACHIEVE low manufacturing costs it is necessary that tolerances be sufficiently large to allow for the dimensional variations that arise from fundamental natural laws that are unchangeable. Tolerances smaller than these will make interchangeable manufacture impossible and thereby result in greatly increased costs and will often require the scrapping of many valuable parts.

Of the infinite number of reasons why perfect parts can not be made, the two most frequently encountered are thermal expansion and deflection or deformation. Thermal expansion is possibly the more important of these two. In the table below, the expansion coefficients, rounded off to six decimal places, are given for some commonly used metals:

	In. per in. per deg. F.
Carbon steel	0.000006
Brass and stainless steel.	0.000009
Aluminum alloy	0.000012
Magnesium	0.000015

It is perhaps better to speak of tolerance in terms of percentage rather than dimensions. Expressing an expansion of 0.0001 in. per in. as 0.01 percent of 1 inch is more significant in that it can be applied to any length. Thus, 0.01 percent means 0.0001 in. in one inch of length; 0.001 in. in ten inches of length and 0.010 in. in 100 in. of length. This last figure (0.010 in.) is the understood tolerance for decimal dimensions when no tolerance is specified on aircraft drawings. It may be readily seen that positioning holes, for example, 100 in. apart cannot be done consistently to this tolerance, even in steel under normal shop conditions.

The process of machining develops thermal expansions which materially affect the dimensional variations in the finished pieces. Some of the causes for these variations are:

1. Temperature of the stock.
2. Temperature of the measuring devices such as micrometers and calipers.
3. Temperature of the machine.
4. Temperature of the coolant.
5. Amount of energy converted into heat by the cutting tool while doing the work.

This last item will affect all of the four preceding items, with a possible exception of the second one. All of these conditions are significant and in themselves, without considering elastic deformation, backlash, and many other factors, make it impractical to meet exceptionally close tolerances on general work.

If a graph of deformations versus lengths be plotted to log scales, the percentage error curves will be obtained as in Fig. 1. The line marked 0.05 percent represents about the most practical error limit under ordinary shop conditions.

The entire question of close tolerances, and to a certain extent, thermal variations, is tied up with the size of the work and the size of the machine involved. Big machines and large pieces of work are subject to greater variations than smaller ones. Obviously, the more metal there is to expand, the larger the change. The dimensions indicated in circuit *C* of Fig. 2 illustrate this point.

When the part being machined is of a metal different from that of the machines or jigs, greater variations will be caused by temperature changes. For example, when machining an aluminum alloy part in a steel jig that is accurate at, say, 70 deg. F., two holes drilled *D* inches apart will be accurately spaced if the temperature of both the part and the jig is 70 deg. F. But if these two holes are to be 100 in. apart in the dural structure and the drilling is done at 50 deg. F. the steel jig will have contracted $0.000006 \times 100 \times (70 - 50$ deg.) or 0.012 in. The dural will have contracted $0.000012 \times 100 \times (70 - 50$ deg.) $= 0.024$ in. If these two holes in the dural structure are to match those in a steel piece drilled in the same jig at the same temperature, the holes will match when both pieces are at 50 deg. F. But if these parts are assembled at 90 deg. F., the steel piece will have a variation of $0.000006 \times 100 \times (90 - 50$ deg.) $= 0.024$ in., and the dural will have a variation of $0.000012 \times 100 \times (90 - 50$ deg.) $= 0.048$ in. Thus, the fallacy of specifying $D = 100.00 \pm 0.01$ in. on the drawing is apparent.

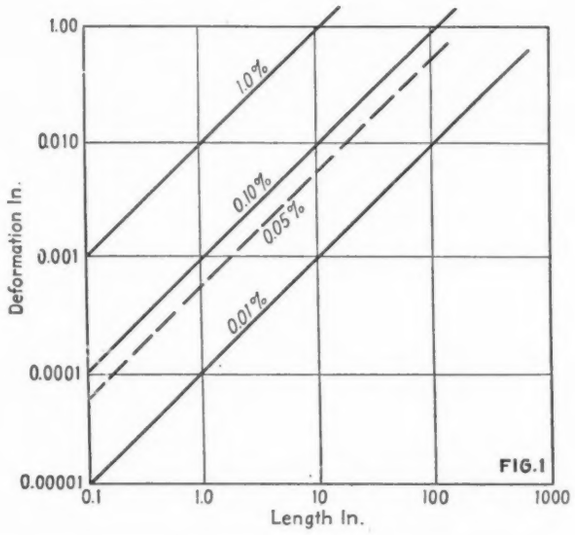
Thermal expansion, as shown above, results in many assembly difficulties. One part may have been made in a machine, such as a spar cap mill, where a good deal of power is used and the coolant becomes lukewarm. Consequently, it expands the metal and when this part is assembled, it may need a large tolerance. This difficulty may be avoided by two means, (1) slow cutting, and (2) by cooling the coolant. It is better, however, not to expect the factory to go to such refinements, as neither method is entirely practical nor consistent with mass production. It is far better to get around these difficulties by relaxing the tolerances. Fig. 3 illustrates an example of a joggled rib attaching to a spar cap. The spar cap was machined to a nominal tolerance of 0.010 in. per cut. The set-up was worked out so that it would be the sum of three tolerances and therefore was 0.030 in. off. The cure was to apply a relaxation. A separate part was made which took care of the variation.

The one factor of thermal expansion has thus far been discussed. Elastic deformation and its effect upon manufacturing and machine operation may now be considered as the second reason why perfect parts cannot be made. In general, errors due to the elasticity of materials may be put in two categories, (1) deflections and (2) manufacturing effects.

In discussing the first subhead, deflection, it is necessary to go into the subject of modulus of elasticity for the commonly used aircraft metals. These are the values of "E" (Young's Modulus) in round numbers:

Magnesium	6,600,000
Aluminum Alloy ...	10,000,000
Steel	30,000,000

By the definition of Young's modulus, it



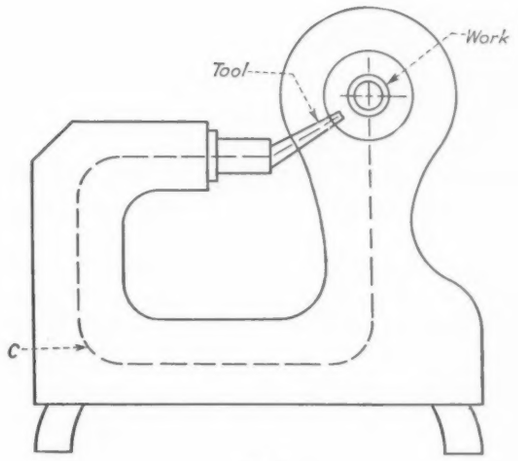
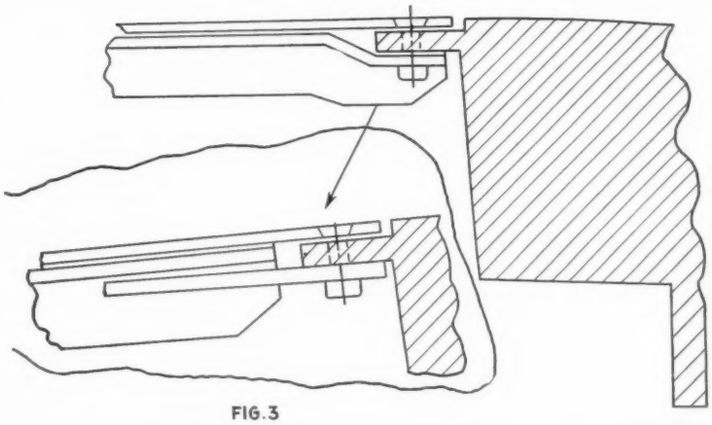


Fig. 1—Percentage error curves obtained when deformations versus lengths are plotted to log scales. Fig. 2—Circuit C indicates the dimensions affected by thermal expansions. The more metal there is to expand, the larger the change.



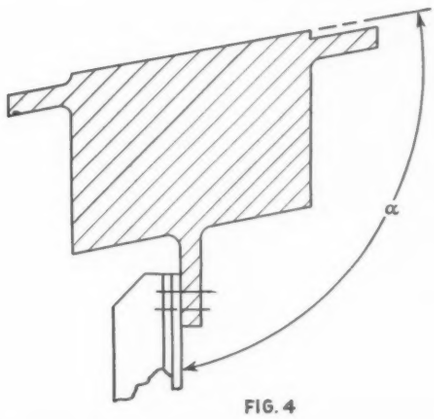


Fig. 3—An example of a joggled rib attaching to a spar cap. Fig. 4—Location of angle alpha on a spar cap.

is the force required to double the length of a piece of material one unit in cross-sectional area. Therefore the ratio of deformation to original length caused by a unit stress will be the stress divided by the value of *E*. Multiplying by 100 gives the percent change in length. In the instance of the aluminum alloy part having the holes 100 in. apart, to obtain an accuracy of 0.01 percent, the dimensions must be held to 0.01 inch. The absurdity of this may readily be seen when it is considered that merely drilling the wing in one position and turning it over would change the dimensions by more than this.

If parts are to be made interchangeable, then care must be taken that the elastic deflections mentioned above and the thermal effects discussed earlier, do not produce distortions greater than those tolerable on assembly. It is almost impossible to meet the standard tolerances for fit of bolts on wing cover panels over any considerable length. The above mentioned elastic effect alone indicates that this would be difficult.

Consider the above three materials of construction in relation to working stresses. It should be remembered that steel, since it is 2.8 times as heavy as aluminum alloy, must work at a stress about 3 times that of the latter to be economical in aircraft. Roughly, the same ratio as the values of *E*, above, should apply here. Thus, working stress should be about:

	Lb. per sq. in.
Magnesium	33,000
Aluminum alloy	50,000
Steel	150,000

As designs improve, higher working stresses can be used more consistently. Greater economy of material and consequent reduction in weight results. Young's Modulus, of course, cannot be changed. However, higher working stresses can be approached by better design.

The second subhead, manufacturing effects, plays an important part in the reasons why perfection of parts cannot be realized.

Consider first a rivet. It is relatively poorly manufactured since it is made in large quantities. Close tolerances in rivet manufacture are unnecessary, since upon being driven, a rivet swells and fills the hole. In swelling, however, the rivet distorts the parts and causes much trouble in fabrication.

Spar caps are critical as to angle α as in Fig. 4. The angle is dimensioned on the drawing to plus or minus 1/10 degree. The rows of rivets, however, may, because of their plastic working, change the angle

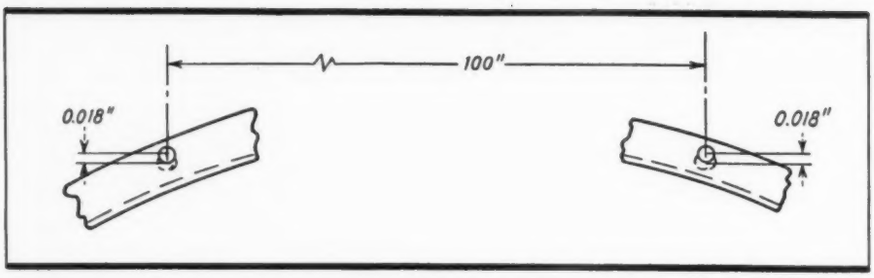


Fig. 5—Attaching angle with maximum slop assumed in the bolt holes.

from 1 to 2 degrees. The amount of this distortion caused by rivets depends upon how they are driven. Squeezing is the most uniform method available.

If all the distortion caused by rivets were uniform, it could be easily taken care of by an alteration of the jig, but such is not the case. A satisfactory jig must be one made to such dimension that the part, when manufactured in it will be the correct finished size. The term for this procedure is "compensating the tooling." The jig must permit the part to take its natural dimensions or restrain it in such a way that it does not cause damage to the part.

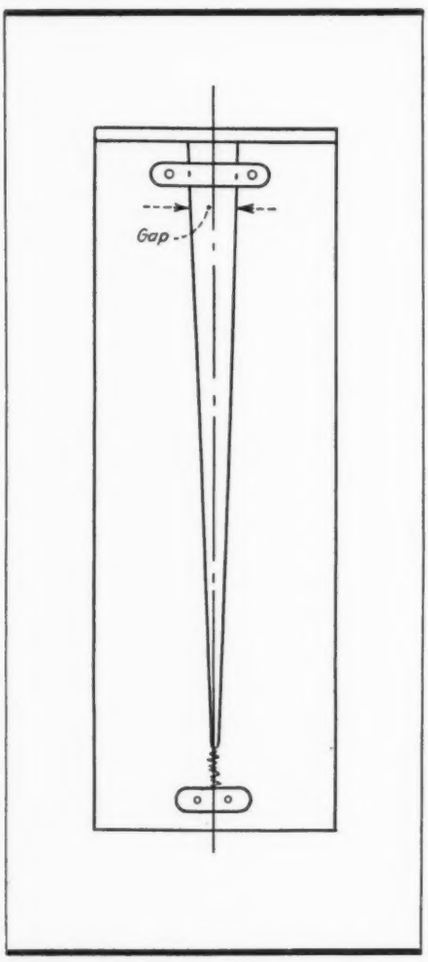


Fig. 6—The jacket of a domestic water heater prior to gas welding.

In aircraft work, the giving of fuselage or wing station numbers to three decimal places implying a tolerance of 0.010 in. causes constant trouble. A station is generally marked, for instance, 235,456 and since the drawing calls for a plus or minus 0.010 in. tolerance, it means that the station, measured from the forward bulkhead or center line of the airplane, cannot vary more than 0.010 in. This is obviously ridiculous. It simply cannot be done. A tolerance of plus or minus 3/8 in. in this instance would be sufficient and would be infinitely more practical.

A particular wing joint furnishes another good example of the impossibility of very close fits under certain conditions. It used to be insisted that the holes for the 1/4 in. attaching bolts be not more than 0.003 in. above nominal size. It was argued that it was unsafe to have greater "slop." Actually, of course, at least half of the holes did not line up and were, therefore, "unsafe" in the terms of the above definition. The holes had to be forced into line with a drift pin. The practical solution of a problem of this kind shown was used in another airplane. Here, the wing attaching bolt holes were made 0.018 in. oversize. It is noted in a design manual, under the tolerance for 1/4 in. hole, that this dimension of 0.018 in. comes in between Clearance Fit (0.028-0.035) and Close Fit (0.001-0.008). The bolt holes, made to this size, were perfectly satisfactory and much less time was expended in the installation. In this attaching angle in Fig. 5, if maximum slop is assumed in the holes at point 100 in. apart, the maximum angle (in radians) will

$$\text{be } \frac{2 \times 0.018}{100} = 0.00036 \text{ radians, or } 0.021$$

degree. The wing will distort to such an extent that a manufacturing tolerance of 0.1 degree is the best that can be expected. Thus, the angular error permitted by the oversize holes is negligible.

As riveting and bolting produces distortions, so also are variations induced when structures are welded. It is easy to see why this is so. The parts are joined while in the molten or near-molten state and when cooling takes place, they shrink, especially in the direction in which the weld was made. Fig. 6 illustrates the gas welding of the jacket of an ordinary domestic water heater. Starting at one end, it is necessary to allow approximately a 2 in. gap in a length of 40 to 45 in. The gap gradually closes up as the weld progresses.

With a recompression (in the recompression type of spotwelder the spots are forged after they are formed) type of spot welding machine, it is theoretically possible to produce spot welds with no distortion. In practice, however, spot welds either shrink, if there is not enough recompression, or expand if there is too much. The parts usually distort about as much as do comparable riveted assemblies.

Machining relieves "locked-in" stresses, but these so-called locked-in stresses in material are not necessarily undesirable. If a long bar of metal is heat-treated, the heat-treat condition on the surface is different from that inside since cooling progresses from the outside surface. The outside contracts and since it is stronger than the warm inside layers, it shrinks and compresses the material while it is still hot. The core material, when it cools, shrinks, and results in a state of tensile stress which tends to compress the outside. The outside of the material is therefore in compression and the inside in tension. The stress pattern of this condition may be somewhat as indicated in Fig. 7.

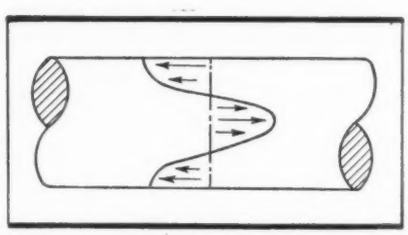


Fig. 7—The stress pattern of a machined heat-treated bar.

If this outside layer is machined off, as is done, for instance, with a spar cap, it allows the inner tension material to react. The result is a shortening of the piece. The amount of shortening can be as much as 0.001 in. per inch of length, plus or minus a variation of 20 percent. Note that this variation is plus or minus 0.002 in. per inch and is uncontrollable. Any heat-treated bar, when machined, will react in this manner.

In summation, it becomes apparent that a practical approach to this question of tolerances is necessary. A close tolerance should not be put on a drawing unless it is essential that it be there. Tolerances specified excessively smaller than the working conditions warrant, or specified with no regard for the limitations of the manufacturing processes used, will result in increased costs and necessitate the scrapping of many parts. The intrinsic nature of engineering materials causes these troubles. They are not the fault of the shop.

Motor Ventilation Requirements For Aircraft Service

L. W. WIGHTMAN
Fractional Horsepower Motor Engineering Division, General Electric Company

Discussion of the engineering compromises involved in the design of small, lightweight, aircraft motors to compensate for the effects of duty cycle, low ambient temperatures at high altitudes, and quantity and velocity of air flow. Different methods of motor ventilation are shown.

IN PREWAR STANDARD fractional horsepower motors high efficiency was considered as the easily measured criterion of over-all motor performance, but the recent need for powerful, lightweight motors, such as used in aircraft, imposes many new engineering considerations.

To get high efficiencies in conventional general purpose motors, the flux density in the iron and the current density in the copper are kept low. To lower these densities, more electrical iron and copper is required and hence greater weight and space is required per horsepower output. Therefore, heat losses to be dissipated per unit of weight or space are low in such relatively large heavy motors and can be easily dissipated by natural convection and radiation without much additional ventilation.

Aircraft motors are different. They are a compromise between the minimum weight and the smallest size. Little importance is placed on efficiency. To obtain the lowest possible weight, the electrical iron and copper is utilized to the fullest extent consistent with safe temperature rise. Because the best possible ventilation system requires considerable space, a compromise is made between increased space for ventilation and motor weight.

Prewar standard fractional horsepower motors also were designed for standardized production. Essentially the same set of parts were used for a cushion or welded base-mounting, to be installed upright, side wall, or ceiling; or the motors could be end flange-mounted. In addition, the motors could be connected electrically for reversible rotation, 115 volts, 230 volts, 50 or 60 cycle operation.

At first it was thought that a standard line of aircraft motors could be designed so the proper type and rating for most applications could be picked from a catalog. However, experience has proved that almost every application required a special motor for each application. These include turret motors and amplidynes, gun-charger motors, wastegate motors, aerial-reel motors, landing-gear motors, and cowl-flap motors. Almost every motor in an airplane is entirely different from every other motor in the craft, not only in rating but in fundamental design.

Two Basic Classes

Aircraft motor applications are divided into two basic classes: Intermittent duty and continuous duty. Most applications require intermittent duty, the motor being selected on the basis of duty cycles. The duty cycle is determined on the basis of the average on-time and off-time of the motor and the amount of load during the on-time.

A duty cycle may be continuous or it may be for only a short duration. An example of a continuous-intermittent duty cycle is an oil cooler flap motor which is continually jogged to keep the oil temperature relatively constant. An example of a short-time duty cycle is a wing flap motor which usually operates twice during a flight.

If the on-time of the duty cycle is short enough and the off-time long enough, the motor need not be ventilated. If the on-time is a little longer, or if the off-time is such that the motor does not quite cool to the ambient temperature, a small amount of ventilation may be required. From this condition, the amount of ventilation needed varies until a condition is reached where the on-time is so long or the off-time so short that a continuous rated motor is required. Here the application may be classed as intermittent, but the motor may as well be rated continuous. These specialized ventilation requirements are one of the reasons for the variety of aircraft motor designs.

The relatively few truly continuous duty aircraft applications include: Generators, motor alternators, dynamotors, blower motors, and pump motors.

Before the war little was known about the effect of ambient conditions on an airplane motor in flight. An aircraft motor must operate at sea level air density over an ambient temperature range of from −60 deg. C. to +70 deg. C. Usually the same motor must also operate at an altitude of 40,000 ft. where the air density is about one-fourth of that at sea level and the ambient temperature may be anywhere between −55 deg. C. outside the airplane to 0 deg. C. inside a heated plane.

Several years ago, many engineers designing accessories believed that the motors would get too cold at high altitudes with such low ambient temperatures. Actually, however, care must be taken to prevent motors from becoming too hot at high altitudes. The much lower air density at high altitude correspondingly decreases the weight of air delivered by the constant volume fan operating at a given speed. Hence, the effectiveness of the ventilation system is reduced since it depends on the rate of the weight of cooling air passing through the motor. This applies only to motors cooled mainly by fan ventilation.

Aircraft motors for very intermittent duty, which have little or no ventilation, get colder at high altitudes because they depend almost entirely on radiation and natural convection to dissipate their heat losses. As the altitude increases, the higher loss by radiation is usually the greater part of the heat dissipation.

Motors Specially Designed

Aircraft motors are specially designed to overcome ambient conditions. First, a balance must be obtained between the quantity and velocity of ventilating air, which, for a given speed and fan diameter, are primarily determined by the cross-sectional area of the air flow paths through the motor. As this area is increased, the quantity of air flow is increased; and the air velocity may either increase or decrease slightly, depending on the fan characteristics, until the point of maximum fan efficiency is reached. Beyond this point, the air velocity and air-

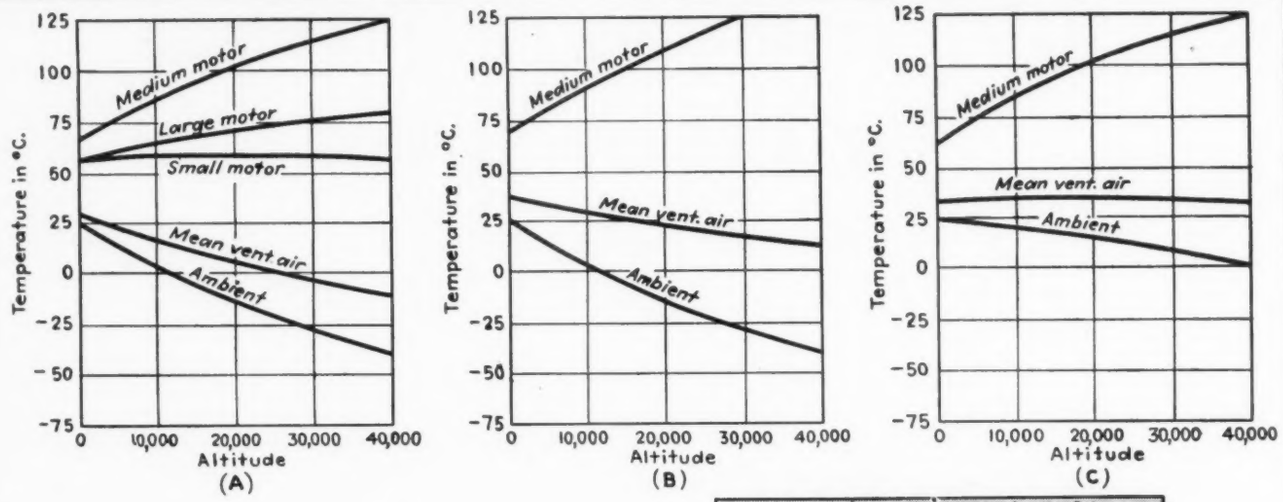


Fig. 1—Aircraft motor temperature characteristics at various altitude. (A) Effect of radiation on small and large motors. (B) Effect of an increased temperature rise of the ventilating air on medium size motor compared with curve in (A). (C) Effect of an increased ambient temperature at high altitude on medium size motor compared with curve in (A).

Approximate Comparison of Aircraft Motor Sizes

Size	Dia. In.		Length, In.		Weight, lb.	
Small	—	3	—	6	0.3	6
Medium	3	5	6	12	6	21
Large	5	up	12	up	21	up

flow resistance decrease rapidly as the area increases and hence the quantity of air flow is increased.

The quantity of ventilating air must be such that the temperature rise of the air as it passes through the unit is not excessive. For highly ventilated aircraft motors that have relatively continuous operation, the temperature rise of the air as it passes through the unit must be held less than that of standard motors. Only in this manner can aircraft motors be designed to take advantage of the low ambient temperature at high altitude.

To keep the air temperature rise down requires an increased quantity of air flow. However, an excess of air flow beyond a reasonable value will reduce the effectiveness of the ventilation. The air velocity determines the effectivenss of surface heat transfer. As the quantity of air flow continues to increase at the expense of the air velocity as a result of increasing the area of the flow channels, a value is reached where the advantage gained by a further decrease in air temperature rise is more than offset by the decrease in the effectiveness with which the heat is transferred from the motor to the ventilating air.

Along with the balance between the quantity and velocity of the ventilating air, a balance must be obtained between the thermal and electrical design as a whole. Any change in the quantity and velocity of the ventilating air through a motor is assumed to be achieved by changing the cross-sectional area of the air flow paths through the electrically active part of the motor. In all a.c. units this is done at the expense of electrical material. In most d.c. units this is also true. But in some d.c. units the required characteristics are such that the lack of interpoles, for example, leaves too much air flow area and resistance is added in the path of the air to reduce the quantity of air flow and increase the velocity.

Since it is generally true that some electrical material must be sacrificed to obtain a balance between the thermal and the electrical design, it is desirable to have a relatively close balance. This is achieved when the desired motor rating is obtained under the worst ambient conditions without excessive motor winding temperatures and at minimum weight within reasonable space limitations.

As explained, the adverse ambient conditions for a highly ventilated aircraft motor occur at high altitudes. The worst condition occurs when the ambient is heated, such as when a motor operates in a heated cabin or near an engine. If the worst high altitude ambient conditions are known, it is possible to determine from normal altitude tests whether the unit will function satisfactorily under the worst high altitude conditions. The effect of the lower ambient temperature at high altitude is much more than offset by the decrease in air density. Therefore at normal altitude, all highly ventilated aircraft motors operate underrated; or else, at high altitude they have a lower rating than at normal altitude.

Typical highly ventilated aircraft motor temperatures for various altitudes are shown in Fig. 1. The divergent shape of the curves shows how the temperature differentials at normal altitude are magnified at

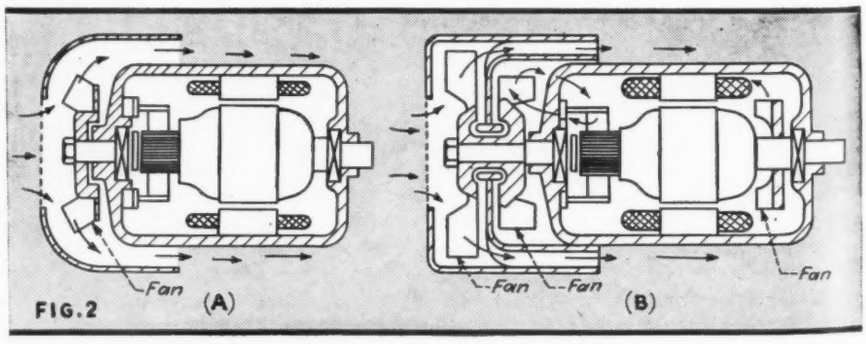


Fig. 2—Two methods of ventilating totally inclosed motors. (A) Air is simply blown over the outside of the motor. (B) Air circulated by a fan at each end of the motor transfers the heat from the commutator and windings to the shell.

high altitudes. The curves in Fig. 1 (A) show the helpful effect of the radiation on small motors as compared with large motors; and also the required under-rated condition of a medium sized motor at normal altitude to keep it within a reasonable operating temperature at high altitude. In Fig. 1(*B*) is shown the effect of an increase in temperature rise of the ventilating air. This effect is much more pronounced when induced draft ventilation is used, because the fan works on the air after it is heated. Since the fan delivers a constant volume of air, the more the air is heated the less is the weight of the cooling air. In Fig. 1(*C*) is shown the effect of an increase in ambient temperature at high altitude. To stay within a reasonable temperature rise at high altitude, a further reduction of normal altitude rating is required when the ambient temperature is increased at high altitude.

Methods of Cooling

Various ventilating schemes that can be used are shown in Figs. 2, 3, and 4. For example, Fig. 2 shows two schemes for ventilating totally enclosed or explosion proof motors. In Fig. 2(A), air is simply blown over the outside of the motor. This scheme is simple, but the heat generated on the commutator and armature windings is not readily transferred to the motor shell where it can be dissipated to the outside air. In Fig. 2(*B*), the heat generated on the commutator and in the armature windings is transferred to the outside shell by a fan in each end of the motor. This scheme, although it is much better than 2(A), is much more complicated. This is particularly true at the commutator end where the internal fan and the external fan are overhung because the brush rigging makes it impractical to place the internal fan inside the bearing.

In Fig. 3 are shown two schemes of forced draft motor ventilation. The only difference is that in Fig. 3(A) a centrifugal fan is used and in Fig. 3(*B*) an axial flow fan is used. The axial flow fan will often shorten overall motor length, but it is not as effective as a correctly designed centrifugal fan system. The forced draft fan of Fig. 3(A) has the advantage of always working in air at the ambient temperature, but it has the disadvantage of a smaller diameter and of excessive loss at its discharge where the air must be made to enter the motor. Any attempt to decrease this loss by vanes or straighteners requires excessive space and complicates the motor construction. All forced draft fans have the disadvantage of having all their losses dissipated directly into the ventilating air, so that the air temperature is increased to some extent before it reaches the electrically active material.

In Fig. 4 are two schemes of induced draft motor ventilation. The scheme shown in Fig. 4(A) can be made very compact, but it has many disadvantages. While air is shown entering the ventilated box-type brushholders, the cooling effect of this air is practically useless because the air flow through the brushholders is in series with the air flow through the spaces between the field windings and over the armature. Also, the fan diameter in this system is generally restricted by the diameter of the through bolt circle.

The scheme shown in Fig. 4(*B*) has the advantage of a maximum fan diameter and of parallel air flow through the brush boxes and the spaces between the field windings and over the armature. This scheme is the best for ventilating highly rated continuous motors which must operate at high altitude. This is because only this or similar schemes offer a maximum effectiveness of the brush ventilation with a minimum effect on the ventilation of the rest of the motor.

All induced draft fans have the disadvantage of working in air which is above the ambient temperature. However, they have an advantage in that all their losses are dissipated into the exhaust air, so that the temperature rise created by the fan losses occur after the ventilating air has passed over the electrically active material.

No air flow through the face of the shaft extension end of the motors is shown in Figs. 2, 3, and 4 because this face is generally used as a mounting surface. Therefore, in air-through ventilated motors, the ventilating air must enter or leave the motor through holes in the periphery of the shaft extension end as shown. Some exceptions to this are dynamotors and two-unit sets such as motor alternators and motor amplidynes. These have a brush rigging on each end, which makes end mounting on either end difficult. Also, the length-over-diameter ratio of many of these units is such that end mounting would not be mechanically practical. Therefore, a central base mounting system is generally used and the ventilating air enters or leaves through the face of the unit opposite the fan end.

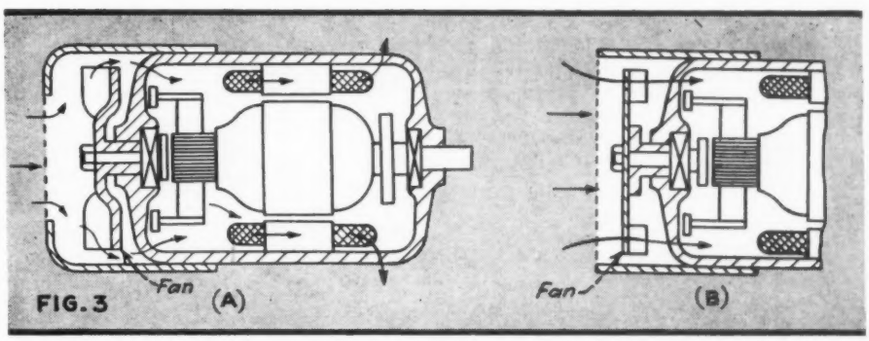


Fig. 3—Two ways of cooling open type motors by forced draft ventilation. (*A*) Centrifugal fan is often more effective. (*B*) Axial flow fan is more compact.

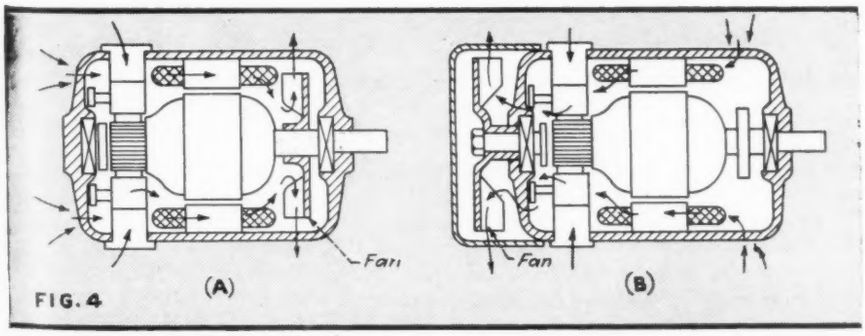


Fig. 4—Two schemes of cooling open type motors by induced draft ventilation. (*A*) Compact, but relatively ineffective because air is heated at commutator end before reaching the field and armature areas. (*B*) Preferred method because it provides maximum fan diameter and parallel air flow.

The general location of a motor may affect the operation of the ventilation system. The location must be such that there is no restriction to the ventilating air as it enters and leaves the motor. This is important and careful designers avoid the natural tendency to tuck motors into convenient corners where the air entrance or exit is restricted, or where the hot exhaust air may be accidentally guided back to the air inlet. Also, motors should not be placed in small or practically enclosed compartments, since the heat generated by the motor losses may raise the motor ambient air temperature above a safe value.

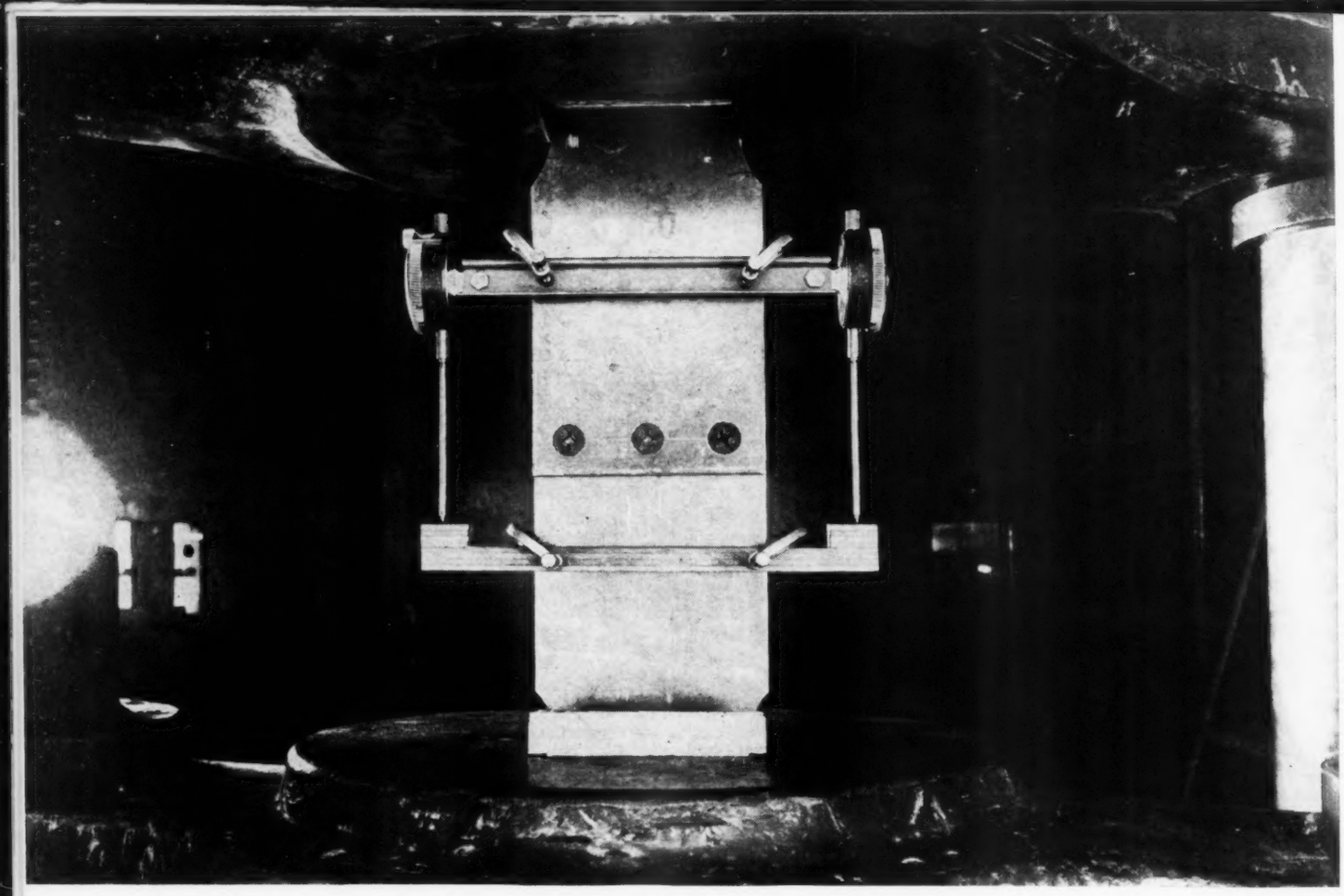


Fig. 1—Shear specimen in hydraulic test machine ready for testing.

Strength of Countersunk Screws In Shear and Tension

HAROLD ADAMS
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Results of an investigation of the strength of 100 deg. and 82 deg. countersunk flat head steel screws in lap joint shear and tension specimens. Allowable shear loads are established from load-deflection curves for both BAC and NAS screws. Results of tension tests are averaged and presented in tabular form.

TO INVESTIGATE THE STRENGTH of steel flat head screws in machine countersunk 24ST aluminum alloy sheet, and to compare the strengths of 100 deg. and 82 deg. head angle screws, lap joint shear specimens and tension specimens were tested to failure.

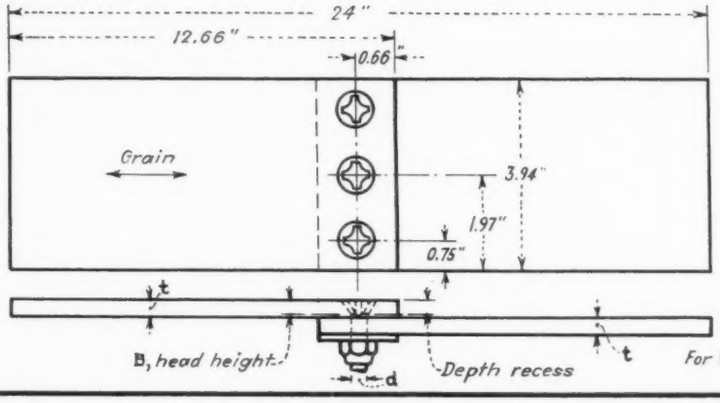
A comparison of the results of the tests on 100 deg. head angle screws with those on 82 deg. head angle screws in machine countersunk sheet indicate:

SHEAR LOADING. A somewhat stiffer joint is provided by 82 deg. head screws in lighter gages where the bearing is critical, whereas the 100 deg. head screws provide stiffer joints in heavier gages.

Except for a critical range where the ratio of screw diameter to sheet gage lies between 2.5 and 4.6, NAS 5/16 in. dia. 100 deg. screws demonstrate greater allowable values than 82 deg. screws. At a screw diameter to sheet gage ratio of 3.0, the maximum strength reduction of 100 deg. screws below that of the 82 deg. screws was 22.5 percent.

Greater allowable loads were demonstrated with NAS 3/8 in. dia., 100 deg. screws than with BAC (Boeing Aircraft Company) 82 deg. screws with ratios of screw diameter to sheet gage less than 2.5, and smaller allowable loads with ratios greater than 2.5. The average reduction of strength of 100 deg. screws below that of 82 deg. screws with ratios ranging from 2.5 to 4.5 was 10 percent.

TENSION LOADING. Ultimate strength of

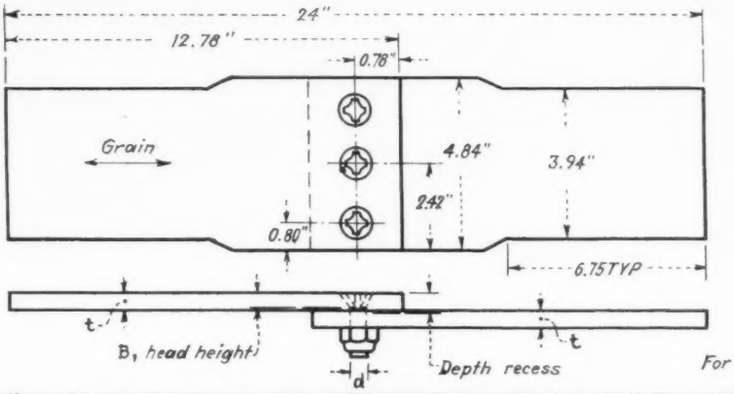


SCREW	t, in.	B/t	Depth recess/t	d/t
NAS 207-16	0.183	0.70	0.87	1.69
BAC 1468-524-14		0.98	1.11	
NAS 207-14	0.162	0.30	0.99	1.91
BAC 1468-524-12		1.10	1.25	
NAS 207-13	0.102	1.26	1.57	3.04
BAC 1468-524-12		1.75	1.99	
NAS 207-12	0.063	2.05	2.54	4.92
BAC 1468-524-10		2.84	3.22	

For NAS 207-All bolts { d = 0.310 in. B = 0.129 in. Depth recess = 0.160 in.

For BAC 1468-524-All bolts { d = 0.310 in. B = 0.179 in. Depth recess 0.203 in.

Fig. 2 (A)—Shear test specimen for BAC and NAS 5/16 in. dia. screws.

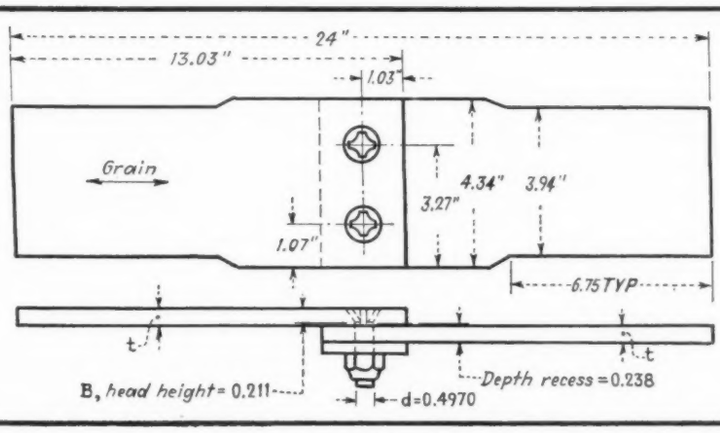


SCREW	t, in.	B/t	Depth recess/t	d/t
NAS 208-9	0.245	0.63	0.76	1.52
BAC 1468-624-16		0.88	0.95	
NAS 208-17	0.183	0.85	1.01	2.04
BAC 1468-624-14	0.190	1.13	1.22	1.96
NAS 208-14	0.127	1.22	1.46	2.93
BAC 1468-624-12		1.69	1.83	
NAS 208-13	0.083	1.87	2.23	4.49
BAC 1468-624-12		2.59	2.80	

For NAS 208-All bolts, { d = 0.3725 in. B = 0.155 in. Depth recess = 0.185 in.

For BAC 1468-624-All bolts, { d = 0.3725 in. B = 0.215 in. Depth recess = 0.232 in.

Fig. 2 (B)—Shear test specimen for BAC and NAS 3/8 in. dia. screws.



SCREW	t, in.	B/t	Depth recess/t	d/t
NAS 210-23	0.310	0.68	0.77	1.60
NAS 210-21	0.245	0.86	0.97	2.03
NAS 210-18	0.162	1.30	1.47	3.07
NAS 210-17	0.102	2.07	2.33	4.87

Fig. 2 (C)—Shear test specimen for NAS 1/2 in. dia. screws.

screws loaded in tension by flat sheets were greater for 100 deg. screws than for 82 deg. screws in every test.

Test Specimens. Sixty lap joint shear specimens were prepared as shown in Fig. 2 (A), (B) and (C). Sixty tension specimens were also made. Identical screw-sheet combinations were used in both shear and tension tests. All screws used were steel heat-treated to 125,000 lb. per sq. in. minimum tensile strength. BAC and NAS 5/16 in. and 3/8 in. dia. steel screws were used in 24ST sheet, Specification AN-A-12, ranging from 0.063 in. to 0.245 in. gage. In this way, a comparison could be made between the NAS 100 deg. head angle screws and the BAC 82 deg. head screws. Additional specimens were made with NAS 1/2 in. dia. screws in 24ST sheet material ranging from 0.102 to 0.310 gage. Two coupons were cut from each sheet of material used in making the test specimens to determine the tensile properties.

Test Procedure. The shear specimens were placed in a hydraulic test machine, as shown in Fig. 1, with a 12 in. length between the grips on the specimen. Deflections of the joint over a six inch span were recorded as indicated by dial gages mounted along each edge of the specimen. Readings were taken at suitable increments as the load was increased continuously to failure, and the ultimate load and type of failure noted.

Tension specimens were mounted in a jig

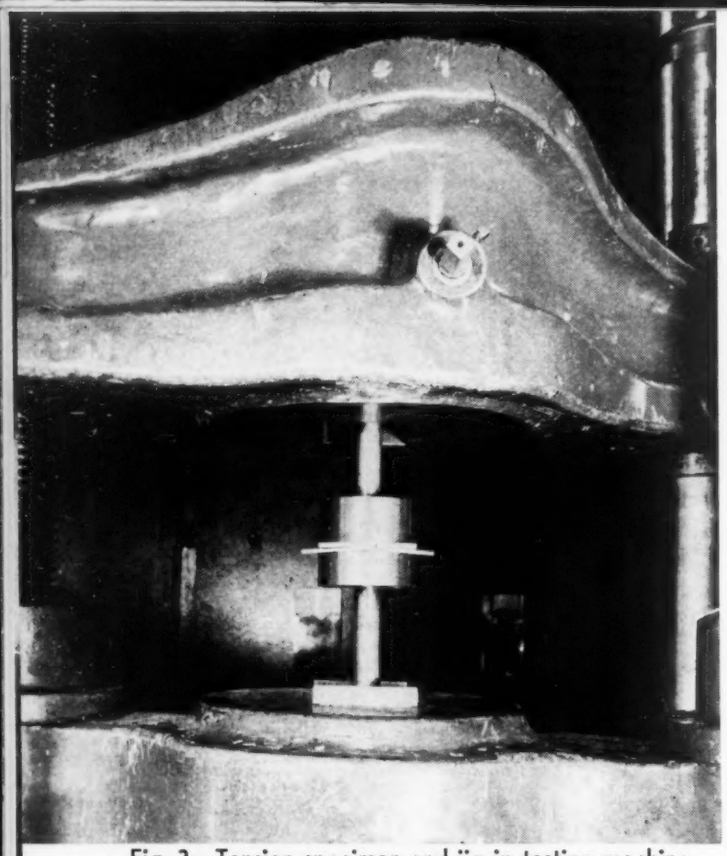


Fig. 3—Tension specimen and jig in testing machine.

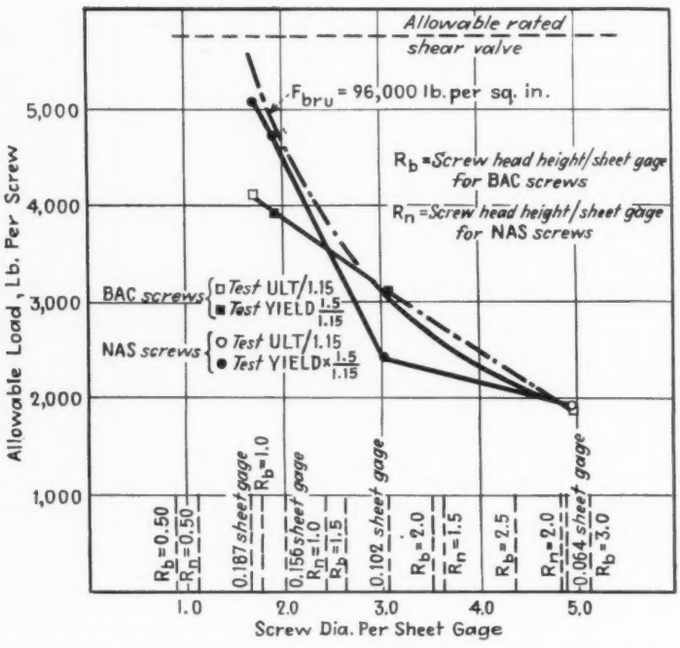


Fig. 4 (A)—Allowable loads with 24ST sheet for 5/16 in. dia. BAC and NAS screws as established by smallest of limiting factors.

Table I—Relative Stiffness of Joints

Screw dia., in.	Sheet gage, in.	Average yield, load × 1.5/1.15, lb.		Average slope, P/δ, 10^4 lb. per in.		Type of Failure
		NAS	BAC	NAS	BAC	
5/16	0.183	5,090	4,260	9.4	6.6	shear
	0.162	4,730	3,935	6.4	9.3	shear
	0.102	2,412	3,120	6.7	11.9	bearing
	0.063	2,190	2,360	6.7	7.0	bearing
3/8	0.245	7,693	7,210	14.5	13.0	brg. and shear
	0.188	6,030	5,635	10.5	11.8	shear
	0.127	4,417	4,828	7.2	12.2	bearing
	0.083	3,490	3,750	6.9	11.7	bearing

installed in the hydraulic test machine and tested to failure. Fig. 3 shows the setup for the tension tests.

Allowable Shear Loads. From the tabulated test results, load versus deflection curves were plotted for all shear specimens. From these curves, a set of curves were plotted, Fig. 4(A), (B), and (C) for allowable shear loads for both BAC and NAS screws. These allowable load values were established by using the smallest of four limiting factors:

1. The average test ultimate load per screw divided by 1.15.

2. The average test yield load per screw times 1.5/1.15. The yield load was taken as the test load at which the deviation from the slope of the straight line portion of the joint deflection curve equalled 2 percent of the nominal diameter of the screw shank.

3. The allowable single shear values for steel bolts heat-treated to 125,000 lb. per sq. in. minimum tensile strength.

4. The sheet material bearing loads determined from BAC allowable bearing stress (96,000 lb. per sq. in.) times actual sheet gage times nominal screw diameter.

To compare the relative stiffness of the joints made with BAC and NAS 5/16 and 3/8 in. dia. screws, Table I was prepared. Table I gives the average yield points and average slopes determined from the load-deflection test curves. The slope data are erratic. But considering the yield values with the trend of the slopes, there is an indication that BAC screws produced a stiffer joint

Table II—Results of Tension Tests

Screw dia., in.	Sheet gage, in.	Average ultimate test load, lb.				Ratio BAC/NAS ave. ult. load, percent	Type of failure
		NAS		BAC			
		No.	Load, lb.	No. 1468	Load, lb.		
5/16	0.183	207-16	5,060	524-14	4,605	91	sheet
5/16	0.162	207-14	4,060	524-12	3,770	92.9	sheet
5/16	0.102	207-13	2,380	524-12	1,900	79.8	sheet
5/16	0.063	207-12	940	524-10	888	94.5	sheet
3/8	0.245	208-19	8,620	624-16	8,210	95.2	*
3/8	0.183	208-17	5,850	624-14	5,772	98.7	sheet
3/8	0.127	208-14	3,550	624-12	3,218	90.6	sheet
3/8	0.083	208-13	1,860	624-12	1,780	95.7	sheet
1/2	0.310	210-23	14,480	.....	.....	....	sheet
1/2	0.245	210-21	10,910	.....	.....	....	sheet
1/2	0.162	210-18	5,837	.....	.....	....	sheet
1/2	0.102	210-17	2,923	.....	.....	....	sheet

*Screw head failed with NAS screws; sheet failed with BAC screws.

than NAS screws where bearing failures occurred, and NAS screws produced a stiffer joint when shear was critical.

Where the ratio of NAS head height to sheet gage ranges from 1.0 to 2.0, the curves of Fig. 4(A), (B) and (C), indicate that the 5/16 and 3/8 in. dia. NAS screws had smaller allowables than BAC screws. The 1/2 in. dia. NAS screws at a ratio of head height to sheet gage equal to 1.3 demonstrated a low allowable.

The shear strengths of the screws were affected by the depth of the Phillips recess in the head. The BAC screws had considerably deeper recesses than the NAS screws as noted in Fig. 2(A) and (B). Consequently, for sheet gages where shear predominated, the area at the plane of shear was reduced by the recess to a greater extent with BAC screws than with NAS screws. This difference is reflected by the resulting reduction of ultimate loads in BAC screws where shear is critical and was noted in the appearance of the screws after failure. Fig. 5 shows a group of five typical bearing and shear failures that occurred in the shear tests.

TENSION TESTS. In the tension tests, the ultimate strengths were greater for NAS 100 deg. head screws than for BAC 82 deg. head screws in every screw-sheet combination. Except for 5/16 in. dia. screws in 0.102 gage sheet, the strength of the BAC screws ranged from 90 to 99 percent of that of the NAS screws. The strength of BAC screws in 0.102 gage sheet was unaccountably low, averaging only 80 percent of that attained with NAS screws.

Sheet failures occurred in all tension tests with the exception of NAS 3/8 in. dia. screws in 0.250 gage sheet, part of the head broke off in this combination. In sheet failures, the sheet deformed sufficiently to allow the screw head to pull out. But the ultimate load was always reached before the head pulled through. There was little noticeable deformation of the screw heads in the sheet failures.

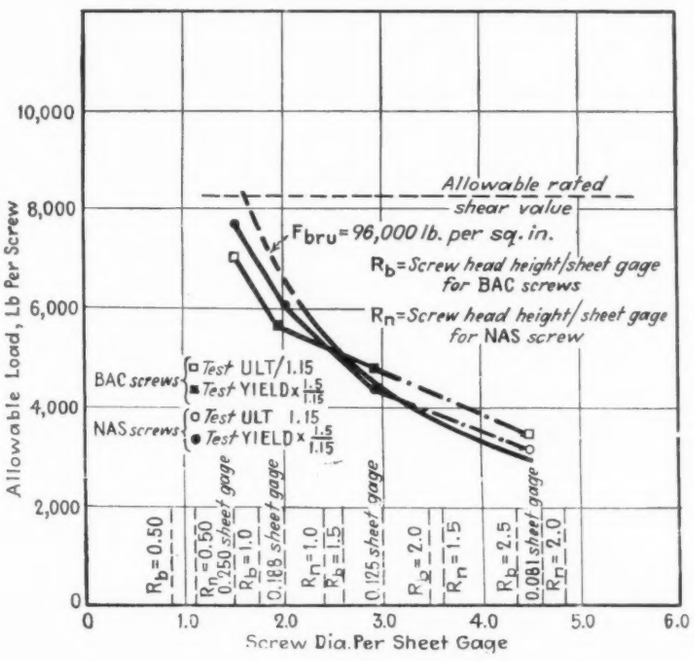


Fig. 4 (B)—Allowable loads with 24ST sheet for 3/8 in. dia. BAC and NAS screws.

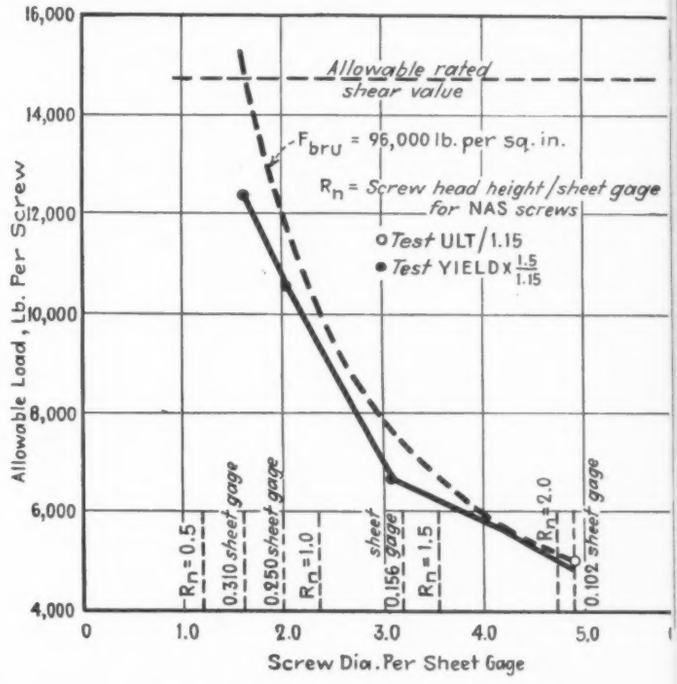
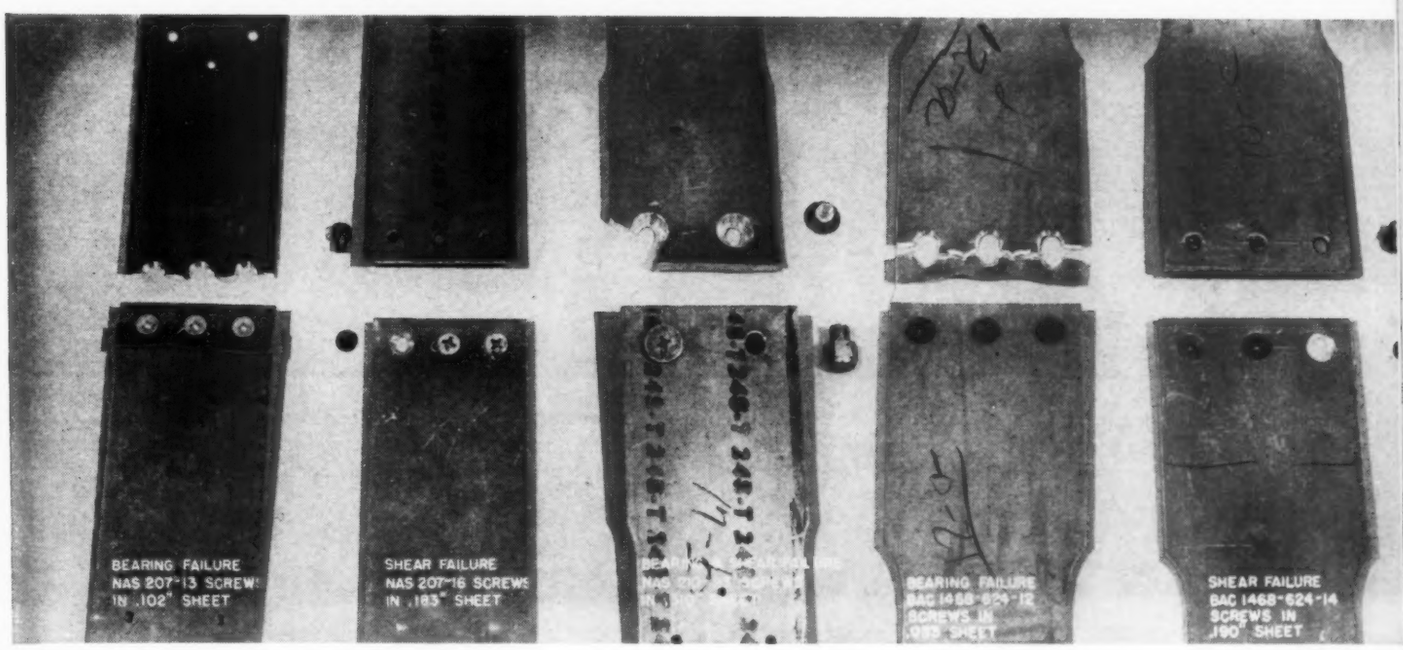


Fig. 4 (C)—Allowable loads with 24ST sheet for 1/2 in. dia. NAS screws.

Fig. 5—Typical failures of BAC and NAS countersunk screws in shear test.


BEARING FAILURE NAS 207-13 SCREWS IN .102" SHEET
SHEAR FAILURE NAS 207-16 SCREWS IN .183" SHEET
BEARING FAILURE BAC 1468-624-12 SCREWS IN .063" SHEET
SHEAR FAILURE BAC 1468-624-14 SCREWS IN .190" SHEET

Requirements and Design Of Automotive Braking Systems

EDMOND SIROKY
Wagner Electric Corporation

Fundamental procedure in proportioning the braking systems of automobile brakes, with an example to illustrate the details, including a discussion of the advantages of four-wheel over two-wheel brakes, the effects of load transfer and charts showing deceleration and stopping distance.

THE PROBLEM of selecting the type, size and number of brakes for an automotive vehicle is often less simple than it may appear at first thought. Broadly speaking, the principal objective is to provide as much braking as possible and yet have the system simple, inexpensive, light in weight, and, above all, dependable and contributive to safety.

The brake designer must also give careful consideration to the possibilities of dissipating the heat generated during brake applications, to the characteristics of the friction or lining materials that are available, and to the fact that no simple system of brakes can give ideal performance under all conditions of load on the vehicle, inclination of the road, and slipperiness of the pavement. Whereas in the early days of the automobile the speeds and traffic density were such that simple systems were quite adequate, the trend has been distinctly toward more and better brakes. This advance in brake performance has naturally been accompanied by greater attention to the details that tend to make the systems conform more nearly to the ideal.

Since the primary function of the brakes is to provide frictional forces at the points of contact of the tires with the roadway, and since in emergency stops these forces should be as large as possible, an ideal braking system under all conditions would provide the maximum friction at each contact point. The normal pressures at the contact points change with the position of the center of gravity of the vehicle, its deceleration, and the inclination of the roadway, see Fig. 1. The frictional force at each wheel is fixed by the torque developed by its brake, except when the tire skids. It follows that the brake torque distribution between front and rear wheels should change with these variables. Thus, during quick stops, on down grades, or when the loading is such that the center of gravity is well forward, the front brakes should provide a greater portion of the total braking.

The first attempt to approach the ideal was the change, years ago, from 2-wheel to 4-wheel braking systems. An example will illustrate the improvement. In Fig. 2, if h is $0.3L$ and the coefficient of tire friction on the roadway is 0.6, then for 2-wheel brakes, (A), the rear-wheel reaction is W_R equals 0.51W and F equals 0.306W maximum. For 4-wheel brakes, W_R equals 0.42W, F_R equals 0.25W, W_F equals 0.58W and F_F equals 0.35W. Under ideal braking distribution the front wheels take 0.35/0.60 or 58 percent and the rear wheels take 0.25/0.60 or 42 percent. The improvement is (0.60–0.306)/0.306 or about 49 percent for the assumed dimensions and low-friction coefficient between tire and roadway.

Had the coefficient been assumed as 1.0, a value representative of dry concrete roads, the improvement would have been about 117 percent but the brake distribution would have been 70 percent at the front wheels and 30 percent at the rear. While these examples show desirable braking ratios for fair and excellent road conditions, it is quite evident that for slippery pavements

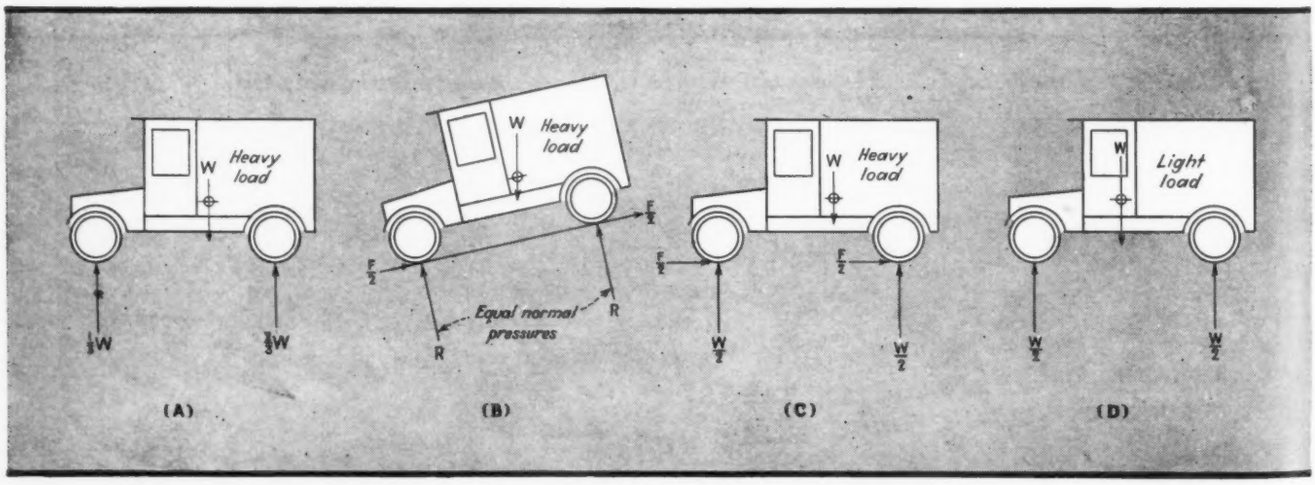


Fig. 1—Changes in axle reactions: *(A)* At rest or traveling at constant speed on level roadway rear axle carries 2/3 load. *(B)* Descending a grade at uniform speed, without brakes dragging, front axle carries more load. *(C)* During deceleration front axle also carries more load. *(D)* Under light loading greater proportion of load is carried by front axle.

NAS 207-13 SCREWS IN .102" SHEET
NAS 207-16 SCREWS IN .183" SHEET
BEARING FAILURE
SHEAR FAILURE
BAC 1468-624-14 SCREWS IN .190" SHEET

covered with ice or snow), the distribution should be nearly that under static conditions. In the example of Fig. 2, this would be 40 percent front and 60 percent rear.

The actual distribution of braking between front and rear axles is generally fixed at a value which the designer deems suitable. It constitutes a compromise between what would be proper for good, dry roadways and that which would exist on slippery pavements. Too much front-wheel braking will cause the front wheels to lock and skid on snow or ice before the rear wheels can develop their maximum friction whereas too much rear-wheel braking will cause the rear wheels to skid during emergency stops on good, dry roadways before the front wheels have attained their maximum effectiveness.

Systems have been devised for altering the distribution of braking effort between front and rear axles to compensate for the effects shown in Fig. 1 but, as might be suspected, the systems are usually quite complicated. One such contrivance, which has been placed in service, is the Wagner "ratio-changer." This is a simple inertia-operated valve so arranged as to decrease the ratio of front to rear braking when operating on slippery pavements. With this device in the system, it is possible to plan on proper distribution under two widely different road-friction conditions and to have nearly the ideal at other times.

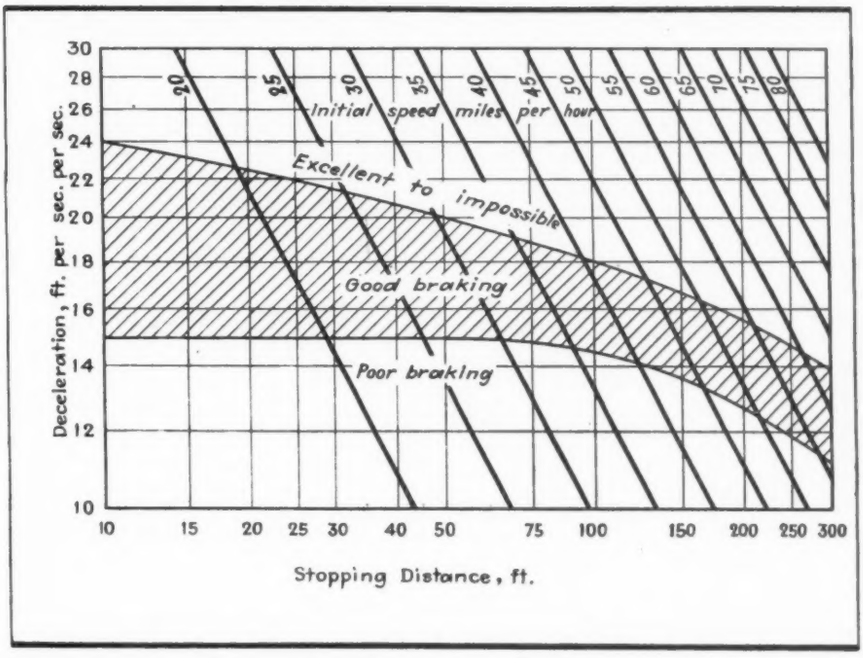


Fig. 3—Stopping distance chart showing the regions of good and poor braking.

Measure of Brake Performance

The braking system of a motor vehicle must perform several functions at different times. It must (1) bring the vehicle to a stop in level driving; (2) slow it and perhaps maintain a reduced speed when traveling down hill; (3) hold the vehicle on up or down grade, either under the effort of the driver or while parked. To provide a standard for comparison, common practice has been to measure brake performance on the basis of ability to make a stop in level driving, either by specifying distances traversed during stopping from different initial speeds or by specifying the average retardations in ft. per sec. per sec. during such stops. For example, if a car traveling at 30 m.p.h. or 44 ft. per sec. were stopped in a distance of 55 feet, its average speed during the stop would be 22 ft. per sec. giving a stopping time of 2.5 seconds and an average deceleration of 44/2.5 or 17.6 ft. per sec. per second..

The average deceleration during a good stop, being nearly the same for different initial speeds, becomes the most useful measure of brake performance. The stopping distance S, in ft. from any initial speed V, in miles per hr. with a deceleration d, in ft. per sec, per sec. can be read from Fig. 3. The formula $S = 1.076\ V^2/d$ can be used in calculating the stopping distance if desired.

Characteristics of Brake-Lining Materials

Practically all modern automotive brakes are fitted with molded asbestos linings, but among them there is a wide difference of properties. For the same pressure against the drum, some linings provide greater friction than others, some suffer more "fading," or loss in friction, during a long application; some show more tendency than others to glaze and lose their frictional values under successive mild applications such as are encountered in continued city driving. Some linings can well withstand contact with high-temperature drums whereas others may suffer loss of friction or may become sticky from "bleeding" of resinous constituents. Others may tend to cause brake noises or may tend to score the drum surfaces. Variations of these properties must therefore be carefully considered when making brake recommendations.

Heat Dissipation From Brakes

Since the function of automotive brakes is to absorb, in friction, the kinetic energy of the moving vehicle and also, in downhill travel, the accompanying change in potential energy, they must in turn dissipate this energy in the form of heat. To facilitate this heat dissipation, it is desirable to provide as much contact area as possible between the lining and the drum, and to have the drum located where air for cooling

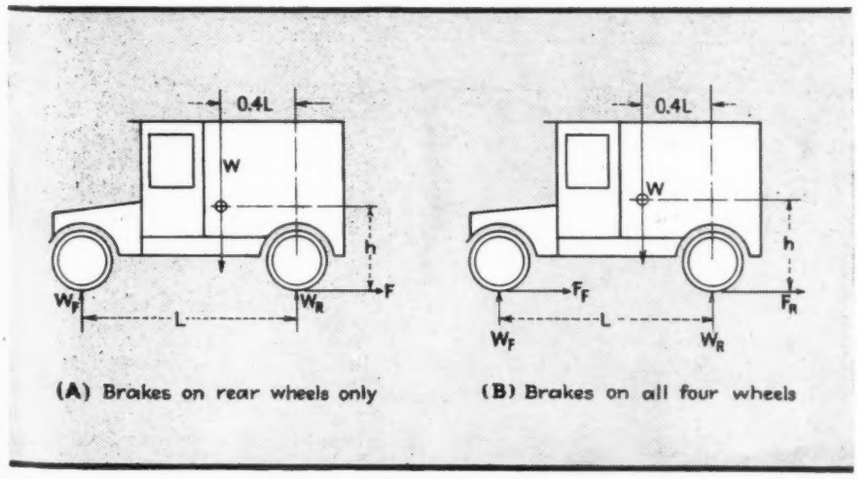


Fig. 2—Wheel loads and braking forces with: *(A)* Brakes on rear wheels only. *(B)* Brakes on all four wheels. Four-wheel brakes are 49 to 117 percent more effective than two-wheel brakes.

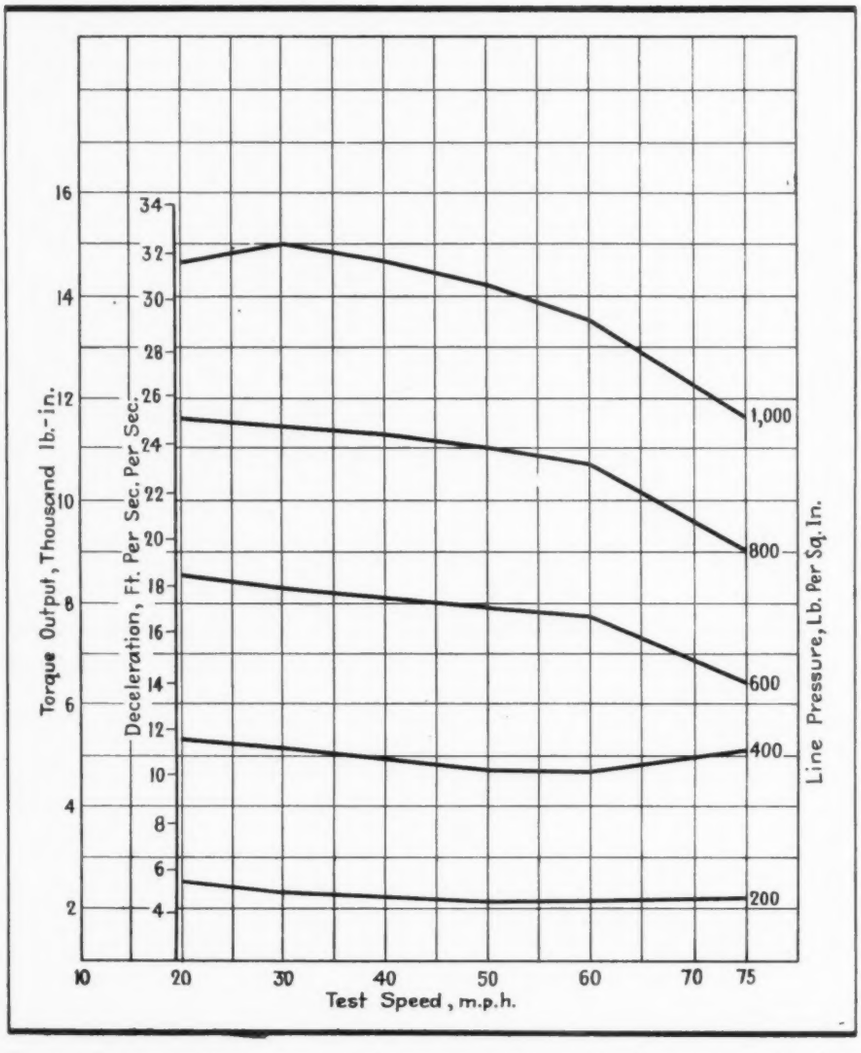


Fig. 4—Dynamometer test curves showing variations of brake torque with vehicular speed and line pressure for Lockheed brake.

Table I—Typical Data from Dynamometer Tests of Automotive Brakes

Vehicle	A	B	C
Type	Passenger Car	Truck	Truck
Brake Size, in	11 x 2 x 3/16	14 x 2 x 1/4	15 x 4 x 3/8
Brake Type	Double-Anchor Lockheed	Double-Anchor Lockheed	Hi-Tork
Wheel Cylinder Diameter, in	1-3/8 to 1	1-3/8	1-1/2 to 3/4
Lining Area, sq. in.	43.4	56.4	120
Tire Radius, in	13.5	18.2	18.2
Maximum Speed, mi. per hr.	75	58	60
Dynamometer Wheel Load, lb.	1,113	2,370	5,740
Axle Weight per sq. in. of Lining	25.7	42.0	47.9
Kinetic Energy per Stop, ft.-lb	208,700	285,200	690,400
Maximum Line Pressure, lb. per sq. in	1,000	1,500	1,500
*Average Torque, Lb.-In	11,300	24,300	50,000
*Average Decel. Produced, ft. per sec. per sec.	24.5	17.5	15.4
*hp. per sq. in. of Lining	1.95	1.80	1.80

*At maximum speed and maximum line pressure, minimum values of five performance tests.

has easy access to it. On the other hand, the dissipated energy must not cause heating of the tire at the rim where vulcanizing can take place. For heavy-duty truck and bus service, the brake diameter is generally made at least 3 in. less than the wheel rim diameter when the brake drum is nested in the wheel. On lighter vehicles so much less energy is absorbed per brake, and the axle construction is so much smaller, that brake drums five to seven in. smaller in diameter than the tire rims have proved quite satisfactory. Undoubtedly large brakes would be desirable on these smaller cars too, from the standpoint of cooling and lining wear, but first costs and unsprung weight have dictated the smaller sizes.

An index of the probable cooling effectiveness of a proposed brake installation is the ratio of the gross weight of the vehicle to the area of lining contact. Although this is not an exact method of determining brake contact area, it has been found quite satisfactory as a general guide, especially since any better method would depend on so many unknown factors, such as the maximum speed at which the vehicle will be driven, whether it will be driven in flat or mountainous country, the frequency with which the vehicle will be stopped, and others. Average values of the gross weight per square in. of brake lining may be taken as follows:

(a) Passenger cars, 22 to 30 lb. per sq. in.

(b) Light trucks and buses, 30 to 40 lb. per sq. in. (Two-shoe, double-anchor brakes, Lockhead type.)

(c) Heavy trucks and buses, 40 to 60 lb. per sq. in. (Hi-Tork brakes only.)

For continuous service in mountainous country the above values should be reduced by about one-third.

Table I has been prepared to show some typical data which have been obtained from dynamometer tests of brakes that have proved satisfactory in service. The values of the last line were calculated by dividing the kinetic energy of the rotating mass by the time required to stop and by the horsepower equivalent.

The variations of brake torque with vehicular speed and line pressure for the brake of column A, Table I are shown in Fig. 4. The line pressure was held constant for each curve. Fig. 5 shows the Wagner Hi-Tork brake and Fig. 6 shows the Lockheed type of brake.

Passenger Comfort

In deciding on brake capacities for passenger cars and buses, it is necessary to take into account the fact that discomfort may be experienced by the passengers when the brakes are suddenly applied. Experience shows that decelerations greater than about 16 ft. per sec. per sec. tend to throw passengers out of their seats. When passengers are standing in the aisles of buses, the maxi-

mum deceleration should probably be limited to about 8 to 12 ft. per sec. per second.

Condition of the Pavement

It is to be recalled that the force which actually stops a moving vehicle when the brakes are applied is the frictional force which develops at the contact of the tires with the roadway. The frictional force under any tire is limited to some fraction "f" of the total contact pressure between the tire and the pavement, the value of "f" depending on the type and condition of the pavement. If an attempt is made to get more frictional force, by further pressure on the brake pedal, the wheel will lock and skid, with the friction force at a value which is generally slightly less than when skidding was impending.

The value of "f" is known as the coefficient of friction between tire and road surface. While its value varies from about 0.6 to about 0.9 for gravel and hard-surfaced roadways when dry, it is reduced to 0.2 and sometimes even less for mud or snow-covered pavements and to about 0.1 or less when ice-covered. Fortunately, the retardation that may be expected from an average dry pavement corresponds quite closely to the maximum retardation consistent with seated-passenger comfort.

Calculating Brake Sizes

To recommend brakes for a given vehicle, procedure as follows has been satisfactory:

1. Select the desired average deceleration, usually 18 to 20 ft. per sec. per sec.

2. Find the percentage of weight transfer for this deceleration from Fig. 7.

3. Select the percentage of weight transfer to be used for establishing the ratio of front-wheel braking ability to rear-wheel braking ability; this is generally taken at one-half to three-fourths of the result in step 2.

4. Compute the effective weight distribution between front and rear axles by adding the weight transfer to the static front-axle load.

5. Calculate the frictional forces on front and on rear wheels by multiplying the transferred loads by the deceleration value from steps (1) and dividing by 32.2 (the gravitational acceleration).

6. Compute the torque by multiplying the fractional force values from step (5) by the rolling radius of the tire.

7. Select suitable combinations of brake diameter and wheel-cylinder diameter from a table of torque outputs of various brakes at the anticipated line pressure.

8. Choose a desirable combination from those selected in step (7) on the basis of cooling effectiveness and axle construction.

9. Determine brake width necessary to

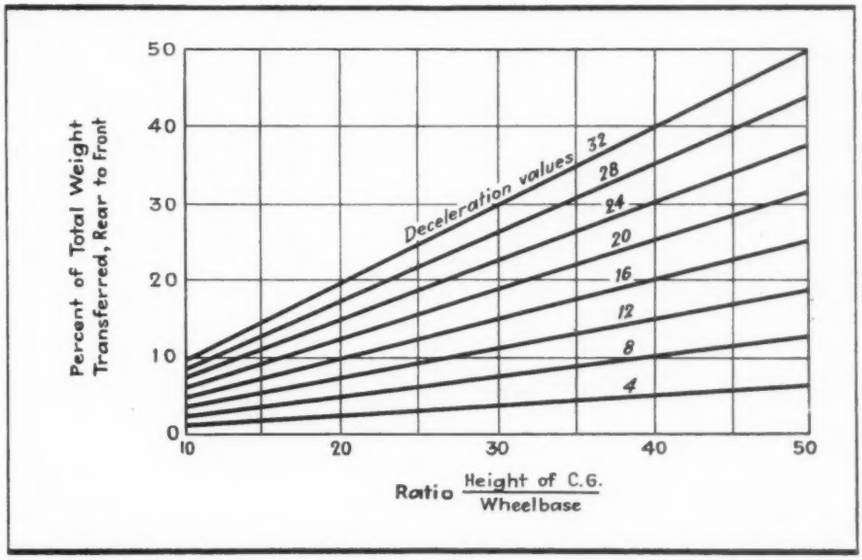


Fig. 7—Weight-transfer chart for determining the percent of total weight shifted from rear to front axle when the brakes are applied.

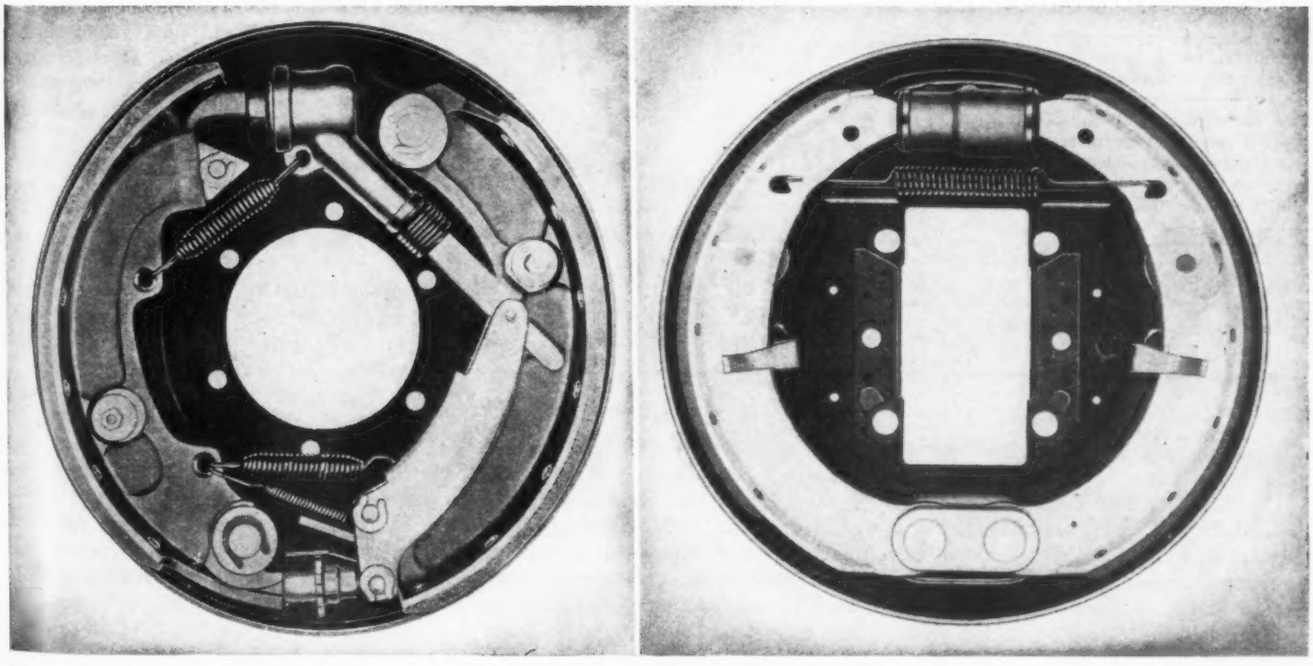


Fig. 5 (Left) A two-forward-shoe brake, Wagner Hi-Tork. Both shoes are self-energized during stoppage of forward motion of the vehicle. Fig. 6 (Right)—Double-Anchor Lockheed brakes are widely used on automobile passenger cars.

Table II—Calculations in Proportioning a Braking System for a Three-Ton Truck

Quantity and conditions	Numerical factors	Value	Units
Total torque required, standard load, 18 ft. per sec. per sec.	18.7 x 18,500 x 18/32.2	193,500	lb.-in.
Desirable front-axle torque 10 percent transfer	0.343 x 193,500	66,300	lb.-in.
Desirable rear-axle torque, 10 percent transfer		127,200	lb.-in.
Front-wheel brakes, double-anchor Lockheed		16 x 2-1/4 x 1/4	in.
Front-wheel, wheel-cylinder diameter		1-1/2-1-1/4	in.
Rear wheel brakes, Hi-Tork		16 x 4 x 3/8	in.
Rear-wheels, wheel-cylinder diameter		1-3/4-7/8	in.
Output, one front brake, 1,500 lb. per sq. in. from 30 mi. per hr. tests		45,000	lb.-in.
Output, one rear brake, 1,500 lb. per sq. in. from 30 mi. per hr. tests		85,000	lb.-in.
Distribution afforded by above, Front		34.6	percent
Distribution afforded by above, rear		65.4	percent
Line pressure required, standard load, 18 ft. per sec. per sec. 30 miles per hr.	$\frac{1,500 \times 193,000}{2 \times 130,000}$	1,115	lb. per sq. in.
Line pressure, overloaded		1,265	lb. per sq. in.
Master cylinder displacement, front axle		1.04	cu. in.
Master cylinder displacement, rear axle		1.57	cu. in.
Master cylinder diplacement, total		2.61	cu. in.
Master cylinder diameter		1-3/4	in.
Master cylinder stroke required, including port coverage	1.09 + 0.08	1.17	in.
Maximum pedal ratio for 8 in. pedal travel (assumed)	8/1.17	6.83	
Line pressure provided by 200 lb. pedal force, 6.5 pedal ratio, 90 percent efficiency, without power	$\frac{200 \times 6.5}{2.4 \times 0.90}$	600	lb. per sq. in.
Lining area, front brakes		132.6	sq. in.
Lining area, rear brakes		270.4	sq. in.
Lining loading, standard load (no transfer), front brakes		33.9	lb. per sq. in.
Lining loading, standard load (no transfer), rear brakes		51.8	lb. per sq. in.

maintain the desirable vehicle load per sq. in. of brake lining.

The master cylinder diameter and stroke required for the vehicle are calculated by a complicated formula, which takes into consideration original brake-shoe clearance, thermal drum expansion, the setting of parts in and the expansion of the wheel and master cylinders, hose and tubing expansion, and which allows a reserve for lining wear. The master cylinder force is found by multiplyng the area by the contemplated line pressure. The pedal ratio is then selected to give desirable values of pedal travel and pedal pressures. Practical maximum allowable values are 9 in. pedal travel and 240 lb. pedal pressure. If the brake system does not permit of values below these limits, then a power assist device must be employed.

As an example suppose brakes are to be recommended for a three-ton truck, having a gross weight under standard loading of 18,500 lb. and a gross weight when overloaded of 21,000 lb. Let the tire size be 9.00x20 and have a rolling radius of 18.7 in. Assume a speed of 50 mi. per hr. If the load distribution is 4,500 and 14,000 lb. on the front and rear axles, respectively, at standard loading, the corresponding per cents of load distribution are 24.3 and 75.7. The load on the front and rear axles under overload are 5,500 and 15,500 lb. respectively.

A deceleration of 18 ft. per sec. is taken as the maximum and a transfer of load of 10 percent represents an average. From the foregoing data and assumptions the data in Table II can be computed.

The example shows how a braking system can be proportioned for a given vehicle by direct methods. The application of the procedure generally yields satisfactory results, although occasionally actual road tests show the advisability of changing wheel-cylinder sizes so as more nearly to equalize the wear on the forward and rear-wheel brakes. This is to be expected, since the height of the center of gravity of the loaded vehicle is usually not known in advance; and even if it were known for one loading condition, there is no assurance that it would not change within wide limits.

New German Insulating Material

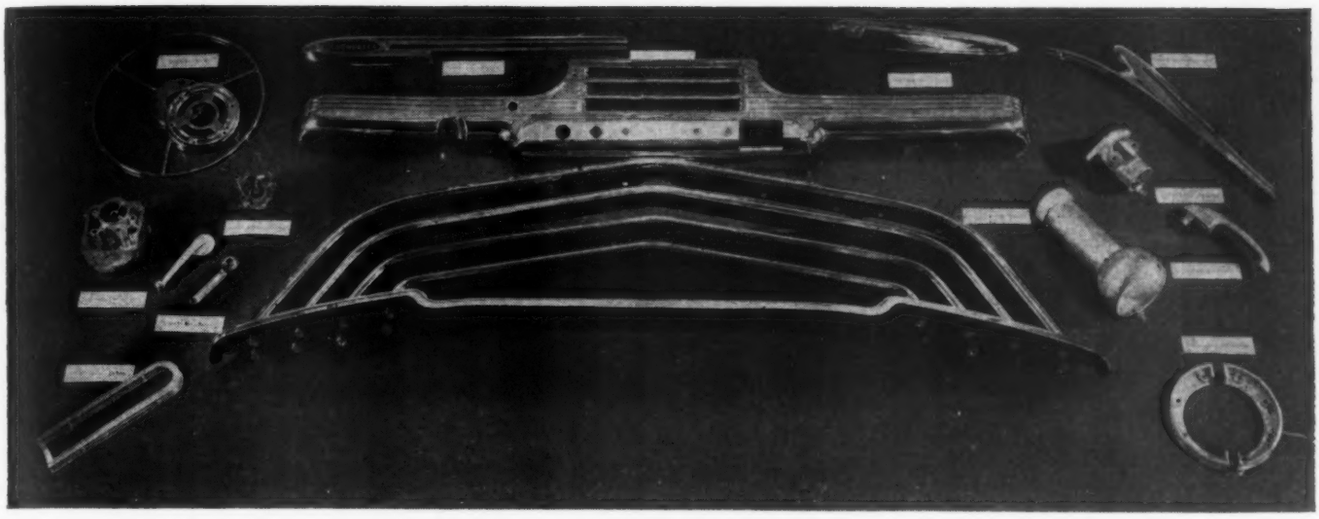
A NEW FOAM MATERIAL, produced in Germany under the trade name "Iporka," is reported by the U. S. Naval Technical Mission in Europe to be an excellent lightweight insulating material for temperatures below 50 deg. C. At higher temperatures, shrinkage is said to occur.

Present uses for "Iporka" in Germany include insulation for icehouses, cold rooms, dry ice containers, and storage cars.

"Iporka" also has potential value for sound insulation, since it reportedly absorbs approximately 50 percent of directed sound.

The new material is prepared by addition of a water soluble ureaformaldehyde condensation product to a foam prepared by vigorous agitation of a water solution of a foaming agent and phosphoric acid. Following further condensation, the resultant foam is dried and cut into desired shapes and size with a knife, saw, or wire.

"Iporka" has a density of 0.9 lb. per cu. ft. and a coefficient of heat conduction of 9.24 B.t.u. per inch per deg. F. at 68 deg. sq. ft. per. hr. mean temperature. In comparison, cork insulation has a density of 10 to 20 lb. per cu. ft. The minimum densities of foams of other resins range from 3.5 to 4.0 pounds per cubic foot.



Representative ornamental and mechanical zinc die castings, most of which are plated.

Applied Finishes For Zinc Alloy Die Castings

A discussion of some organic, plastic, electrolytic and chemical immersion finishes applicable to zinc alloy die castings. Appearance, commercial adaptability and general utility are considered as well as methods of application, limitations and practical uses.

THE FOUR distinct types of applied finishes available for use on zinc alloy die castings include: 1. finishes applied by simple immersion in chemical baths; 2. finishes applied by electrolytic treatment; 3. organic finishes applied by spraying, dipping or brushing; 4. plastic finishes applied by dipping, spraying, or molding.

Certain of these finishes are so important from appearance and sales angles that, if they were not available, the total production of zinc die castings might be half that made today. Nevertheless, millions of die castings have been in use for many years without any applied finish in locations where appearance is of no importance.

Several purely chemical (as distinct from electro-chemical) treatments involving only short dips or spraying in chemical solutions are commercially available. Some of these give the zinc alloy a pearly or opalescent appearance without pronounced change in color, but most of them produce a decided color change ranging from yellowish to olive drab and black.

Most often, these chemical treatments are applied to prepare surfaces for subsequent organic coatings, the adherence of which is improved thereby. But many yield a coating that has in itself a considerable degree of corrosion resistance whether coated subsequently or not. Several inhibit the spread of corrosion when finishes are scratched through or chipped. Some provide colors that are satisfactory for certain uses as final finishes. At least one (Iridite) can be dyed black, red, blue or green. Several are patented products or involve processing patents. Proprietary chemical immersion coatings include:

Cronak (The New Jersey Zinc Company) usually produces a golden brown color showing some iridescence. Is useful with or without subsequent coating.

Iridite (Rheem Research Products, Inc.) usually bronze or olive green but can be dyed black, red, blue or green. Is useful with or without subsequent coating.

Parkerizing (Parker Rust-Proof Co.) dull gray. Usually dyed black, oiled or waxed, also available in standard colors. Useful with or without subsequent coating.

Bonderite (Parker Rust-Proof Co.) (applicable also by spraying) dull gray. Intended for use as undercoat for subsequent organic coating.

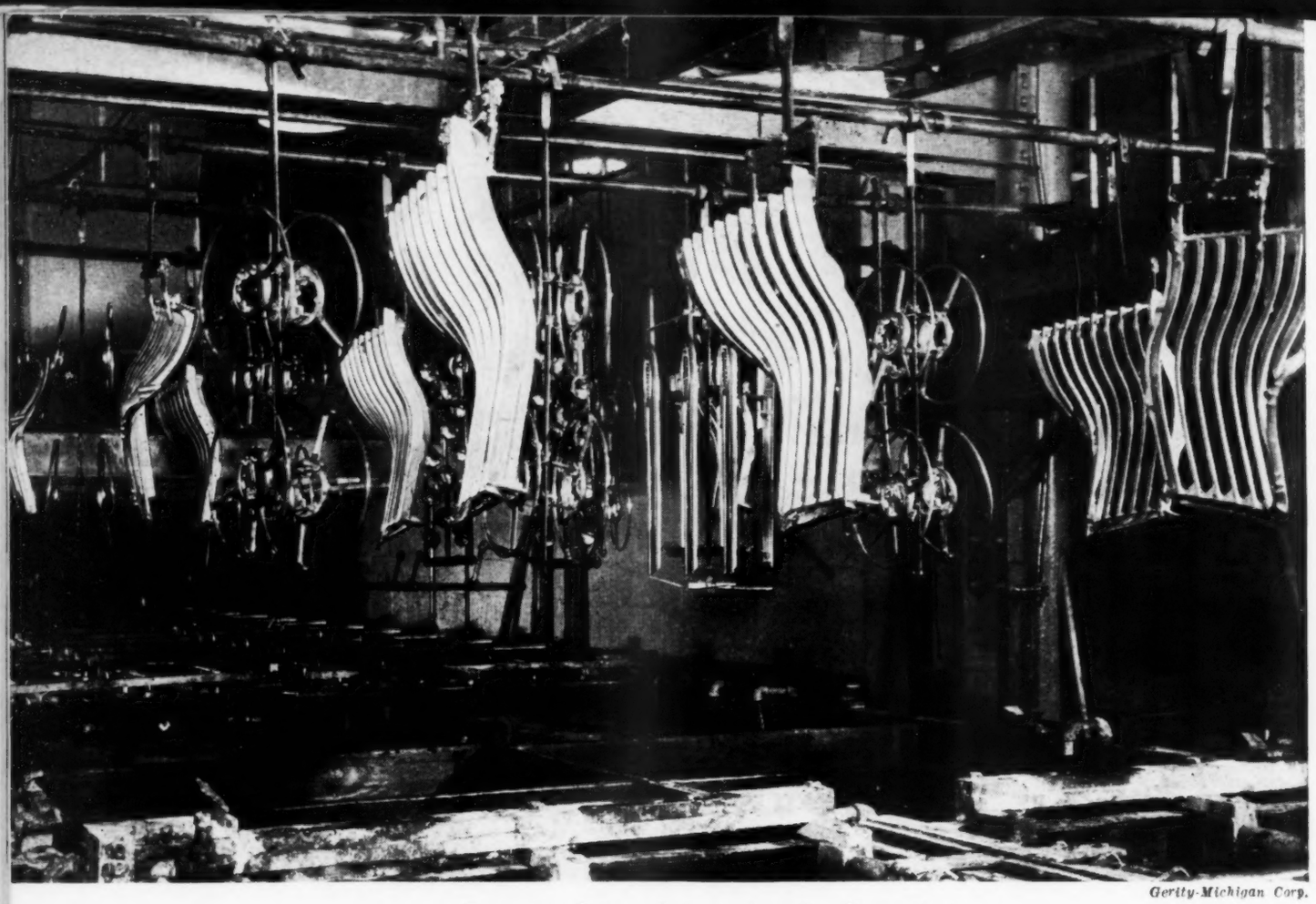
Lithoform (American Chemical Paint Co.) (applicable also by spraying or brushing) dull gray. Intended for use as undercoat for subsequent organic coating.

Ebonol Z (Enthone Co.) jet black. Is suited for indoor exposure or, with hard drying wax or lacquer, for outdoor exposure. Is useful also as undercoat for organic coating.

Black Magic (Mitchell-Bradford Co.) jet black. Is useful with or without subsequent coating. Several non-proprietary solutions containing nickel or arsenic compounds produce black deposits on zinc but some such deposits are not very adherent and require "fixing" with wax or lacquer. Not all are color fast.

Most electro-chemical treatments for zinc are cathodic, but in one, Anozinc (United Chromium Co.), the work is made the anode. This treatment yields either a yellow iridescent, or a black film. The yellow film can be decolorized and is said to produce a "silvery pastel iridescence." Anozinc films are useful also as undercoats for organic finishes.

Black electro deposits can be secured from many solutions. One proprietary type called



Gerity-Michigan Corp.

Conveyor plating of zinc automotive die castings.

"Moly Black" (E. I. duPont de Nemours & Co.) is rapidly deposited but the film deposited is largely non-metallic. Color is deep black and lustrous.

Several of the metals commonly employed in plating, including copper, nickel, cadmium and chromium, can be coated directly on zinc but superior results are secured when the initial coat is copper. Coats of copper and nickel are commonly used under chromium. Heavy coats of copper followed by nickel in about equal thickness and then by a flash of chromium are almost invariably specified for bright exterior hardware, grills and other die cast zinc trim parts for automobiles. Similar specifications, but usually for somewhat lighter coats, are used on hundreds of die cast zinc parts for interior applications.

With plated coatings on zinc (as on most metals), the degree of protection afforded is almost directly proportional to the total thickness of coat. All plated coatings are somewhat porous. If moisture penetrates the coating and stands in contact with the metal of the casting, a white zinc oxide is formed and sometimes leaches through to the surface. This oxide, however, is easily rubbed off and does not stain the plate. With reasonable care, plated coatings of proper thickness correctly applied yield many years of service and retain their brilliance even under severe exposures.

The following minimum thickness (in.) of copper, nickel and chromium plating on significant surfaces of zinc die castings is required under General Motor Specifications:

Class	AA	A	AB
Total	0.0015	0.001	0.00075
Copper	0.0005	0.0003	0.0002
Nickel	0.0005	0.0003	0.0002
Chromium ..	0.00001	0.00001	0.00001
Salt Spray-hr.*	48	32	24

* At 95 deg. F. 5 deg. F. for minor pin holes. Base metal corrosion should not exceed one spot per sq. in. of significant surface. This refers to spots visible to naked eye. No such spot should exceed 1/16 in. in any dimension.

Most plated coatings are dull as they come from the plating bath, but with modern plating solutions and controls, bright copper and bright nickel coats are produced. All chromium plating for decorative purposes is bright. Use of bright plates that require no color buffing save much extra labor and handling, but the solutions used require exacting control.

Some metals, especially copper and nickel, can be applied to small zinc die castings by barrel plating. This saves much labor in handling and racking and, especially for low cost products, has helped to extend the utility of parts of this type.

Some zinc die castings, especially flashlight reflectors, are plated with silver and some, such as razor parts, are gold plated. In both cases, a primary coat of copper is used.

Many zinc die castings, especially for the hardware trade, are plated with brass. This coat sometimes is a yellow or golden color and sometimes a bronze color.

Organic finishes include hundreds of materials or formulations, classed as enamels, varnishes, lacquers and paints, whether clear or pigmented. Refer to "Plastic Resins as Protective Coatings," PRODUCT ENGINEERING, page 1, Jan. 1946. These are available for use on zinc die castings in practically any color and in textures ranging from matt to high gloss. Some are available in forms that yield a "crackle" or leather-like texture that tends to mask minor surface defects of the casting and is in great favor for some types of parts. Many contain synthetic resins and are sometimes referred to as "synthetic" finishes.

Many organic finishes suited for use on other metals are also satisfactory on zinc die castings. To insure proper adherence, however, it is necessary either (1) to use a finish that does not react with or produce a saponifying action on the metal of the casting or (2) to apply first an inhibiting film.

Optimum results are usually secured when the casting (after proper preparation, including cleaning of the surface) is given one of the chemical "pre-treatments" mentioned above before the organic finish is applied. Some makers and many users of organic finishes recommend the phosphate pre-treatment, as it not only promotes good adhesion but inhibits chemical reaction between the zinc and any subsequent coating. Nearly if not quite equal results on both scores are

secured with the chromate pre-treatments and entirely satisfactory results can be had when the proper type of primer, especially of the zinc-chromate type, is applied in place of pre-treatment.

There are enamels available that provide a satisfactory finish for some purposes on zinc die castings in a single spray coat although a more enduring finish and one of better appearance results when more than one coat, of which the first may be a primer, are employed.

Satisfactory results are possible with nitrocellulose lacquer enamels, especially where rapid air drying is desired, but other air drying enamels are also available. Certain types of clear lacquer are sometimes applied over areas of castings that have been buffed (in lieu of plating) when only indoor exposures are made. The lacquer is then used primarily to prevent tarnishing and often does this for a considerable period. It is necessary, however, to select a lacquer that does not react with the zinc as, otherwise, milkiness or poor adherence may result.

With organic materials, the most enduring finishes are commonly attained, (especially for synthetic enamels), when the finish is baked for the time at the temperature specified by the maker. There are, however, enamels designed for air drying that yield finishes suitable for some purposes.

It is important that the times and temperatures given below should not be exceeded in baking as, otherwise, the dimensional stability of the castings may be adversely affected:

Zamak 3 or 5 corresponding to ASTM alloys XXIII and XXV

½ hour	1 hour	2 hours	3 hours
at 425 F.	at 375 F.	at 325 F.	at 300 F.

Composite finishes, such as clear lacquer over silver plating or the application of enamel in recesses or over other areas of plated castings, are frequently employed to prevent tarnishing or to provide pleasing contrasts. In such cases, however, it is important to select a finish that will adhere properly to the usually very smooth surface below it. Lacquers or enamels developed especially for this purpose should be used. Adherence is sometimes improved by baking but the temperature should not be so high as to cause blistering of the plating or to exceed that given in the above table.

Wood-grain finishes, especially on certain areas of plated die castings forming parts of automobile instrument panels, have sometimes been used. Such effects are produced either by a special gelatine process that applies the grain to an enamel coating or by the application of decals that are cemented on. A clear lacquer or varnish is usually applied over such finishes as well as over decals for trade-marks and labels that are sometimes applied to enameled die castings.

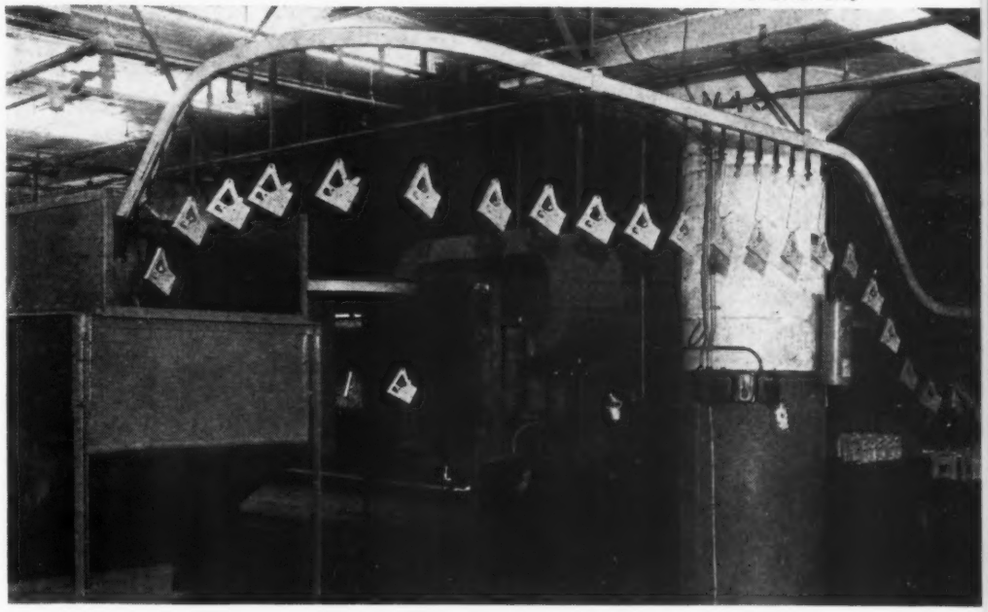
Finishes consisting of a thick coating of plastic that is usually either transparent or translucent are sometimes applied to zinc die castings with pleasing and enduring effects. Such finishes are usually cellulose nitrate, cellulose acetate or ethyl cellulose and should not be confused with the lacquers or enamels that contain considerable percentages of resins.

True plastic finishes are the same as or very similar to plastics used for molding. In fact, one method of application is to injection mold a coating (usually 0.030 in. or more in thickness) around the die casting. This is expensive but has the advantage of providing a part that looks as if it were molded entirely from plastic, yet has the strength and shock resistance of the die casting.

Much more extensively applied, especially on interior hardware parts for Ford cars, is the method, used by Macoid and others, in which the part is repeatedly dipped in or sprayed with the plastic reduced to liquid form. Such coatings are usually built up to a thickness of 0.004 to 0.005 in. or more and possess the natural beauty (sometimes including iridescent or metallic effects) of transparent or translucent plastic.

As these types of finishes have little or no adherence and usually shrink considerably in drying, it is necessary that they form a complete envelope around the part or at least are so applied as to hook around an edge or concealed face. Otherwise, peeling is likely to result. When properly applied, such a finish imparts unusual beauty and possesses exceptionally long life.

Die castings passing through a degreaser on a conveyor.
I. T. E. Corp.



Spray coating zinc die castings for the Link Trainer.
Link Aviation Devices, Inc.

Limit Switching Mechanisms—

Limit switches, used to confine or restrain the travel or rotation of moving parts within certain predetermined points, are actuated by varying methods. Some of these, such as cams, rollers, push-rods, and traveling nuts, are described and illustrated below.

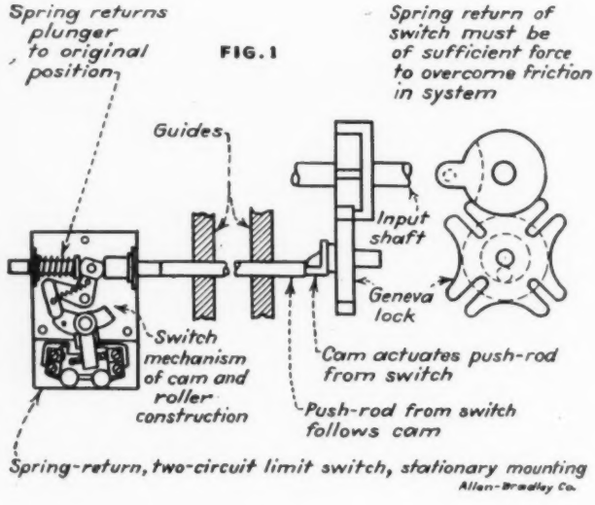

Spring returns plunger to original position
Spring return of switch must be of sufficient force to overcome friction in system
Guides
Input shaft
Geneva lock
Switch mechanism of cam and roller construction
Cam actuates push-rod from switch
Push-rod from switch follows cam

FIG. 1

Spring-return, two-circuit limit switch, stationary mounting

Allen-Bradley Co.

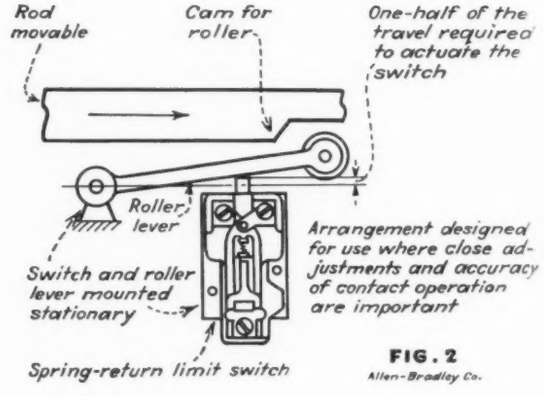

Rod movable
Cam for roller
One-half of the travel required to actuate the switch
Roller lever
Arrangement designed for use where close adjustments and accuracy of contact operation are important
Switch and roller lever mounted stationary
Spring-return limit switch

FIG. 2

Allen-Bradley Co.

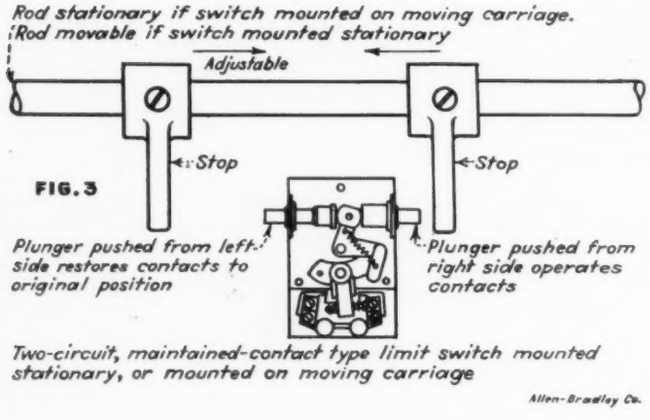
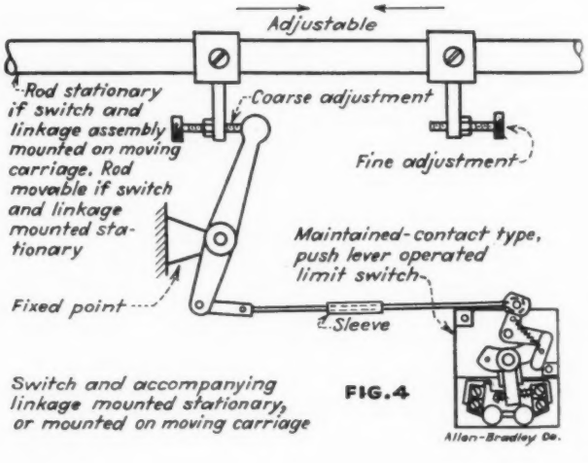

Rod stationary if switch mounted on moving carriage. Rod movable if switch mounted stationary
Adjustable
Stop
Stop
Plunger pushed from left-side restores contacts to original position
Plunger pushed from right side operates contacts

FIG. 3

Two-circuit, maintained-contact type limit switch mounted stationary, or mounted on moving carriage

Allen-Bradley Co.


Adjustable
Rod stationary if switch and linkage assembly mounted on moving carriage. Rod movable if switch and linkage mounted stationary
Coarse adjustment
Fine adjustment
Maintained-contact type, push lever operated limit switch
Fixed point
Sleeve

Switch and accompanying linkage mounted stationary, or mounted on moving carriage

FIG. 4

Allen-Bradley Co.

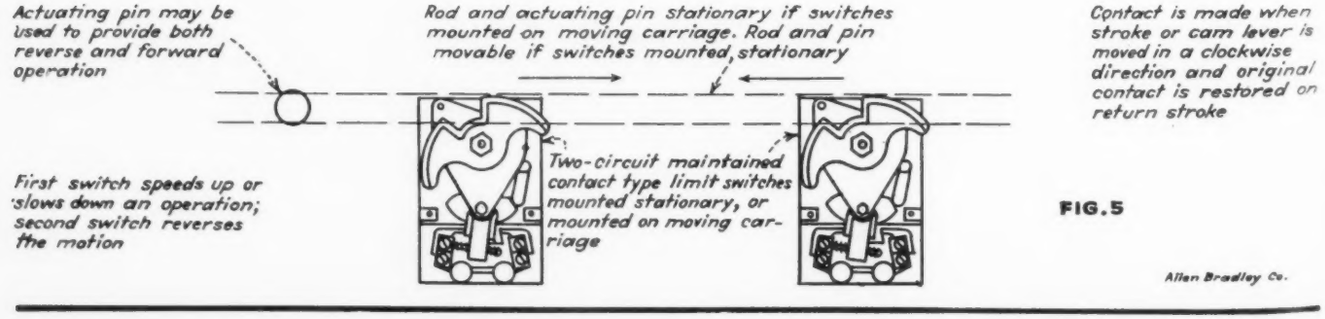

Actuating pin may be used to provide both reverse and forward operation
Rod and actuating pin stationary if switches mounted on moving carriage. Rod and pin movable if switches mounted stationary
Contact is made when stroke or cam lever is moved in a clockwise direction and original contact is restored on return stroke
First switch speeds up or slows down an operation; second switch reverses the motion
Two-circuit maintained contact type limit switches mounted stationary, or mounted on moving carriage

FIG. 5

Allen Bradley Co.

Methods of Mechanical Actuation

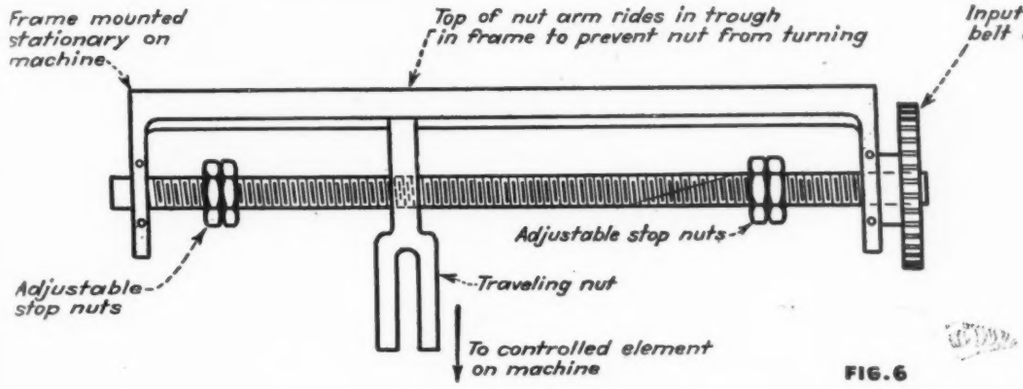

Frame mounted stationary on machine
Top of nut arm rides in trough in frame to prevent nut from turning
Input through gear train, belt or chain drive
Adjustable stop nuts
Traveling nut
Adjustable stop nuts
To controlled element on machine

FIG. 6

Screw and nut device to control travel of machine by reversing belt or throwing out a catch after any specified number of revolutions.
Travel is adjusted by stop nuts

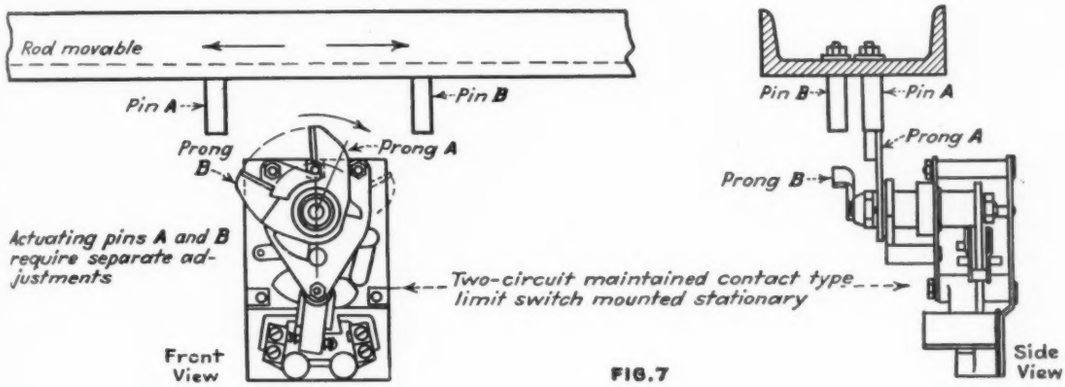
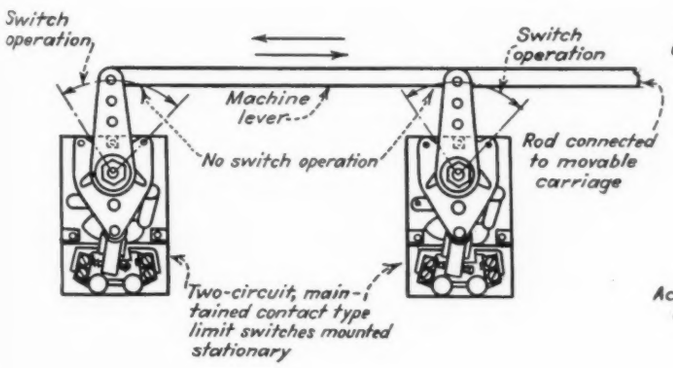


FIG. 7

Contact operation takes place when fork lever is moved. Contacts are restored when fork lever is moved in opposite direction

Allen-Bradley Co.

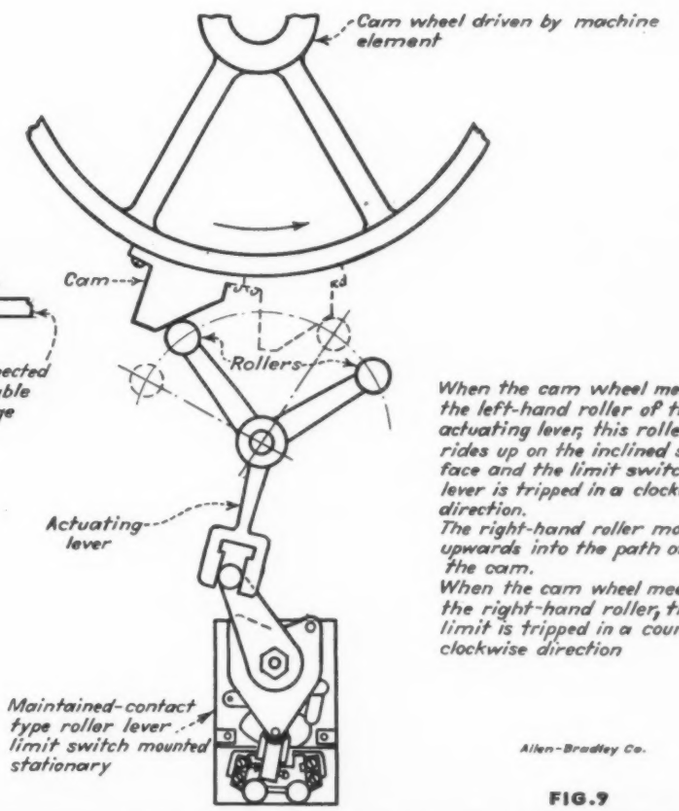
Movement of the machine lever to the right operates the contacts of the right-hand switch, but no contact takes place in the left-hand switch.
Movement to the left operates the contacts in the left-hand switch, but no contact takes place in the right-hand switch



A spring return mechanism can be used if the weight and friction of the connecting linkage does not offset the power of the return spring

FIG. 8

Allen-Bradley Co.



When the cam wheel meets the left-hand roller of the actuating lever, this roller rides up on the inclined surface and the limit switch lever is tripped in a clockwise direction.
The right-hand roller moves upwards into the path of the cam.
When the cam wheel meets the right-hand roller, the limit is tripped in a counter clockwise direction

Allen-Bradley Co.

FIG. 9

Various types of construction, and typical laminates having paper, fabric, wood and aluminum plies.

Physical Properties of Corrugated Low-Density Structures

C. A. SCOGLAND
Plastics Consultant

Laboratory tests on newly developed cellular laminates are reported in condensed form. Methods of sample preparation are discussed. Data presented cover flexural strength, compressive strength, impact strength and heat-transmission characteristics.

RECENTLY, a one-step technique which produces lightweight laminates of cellular construction, to simple or compound curvatures, has been perfected in the Studebaker research laboratories. Laminates of paper, cotton, fabric, glass textiles, aluminum, steel and plywood have been produced with the process. Although a new development, the method opens interesting possibilities for the design engineer. The cellular construction offers great stiffness, fabrics can be laminated to metal to produce panels requiring no surface finish. Since the method is new, only laboratory data are available, and no extensive product testing has been done.

The air-coring in the new laminates is produced with alternate layers of sand-filled tubes placed in the built-up laminate before molding between heated platens in a compression press. The sand-filled tubes are placed so that in each row, each adjacent core is immediately over the void in the row above or below. Consequently, in molding, the sand-filled cores produce a corrugated

structure of substantially trapezoidal cross-section. After molding the sand and cellophane are blown out with an air hose. Because of the flexibility of the sand-filled cellophane tubes, contouring of panels is possible. It is also possible to vary the direction and type of corrugations.

Physical Tests

The samples tested had a total of 32 plies. The surface elements were composed of 10 sheets of resin-impregnated paper and a single sheet of resin-impregnated fabric. The center element contained ten sheets of resin-impregnated creped paper with the crepe running longitudinally with the sand-filled cellophane tubes. Thickness of test piece was ½ in. In the heat transmission test, the samples were tested with air cores vertical, and also with air cores horizontal, and for purposes of comparison a solid sheet of 120 plies with a sheet of fabric on each surface was also tested.

In testing for compressive strength, a cross-head loading speed of 0.025 in. per min. on a Baldwin-Southwark-Emery Universal 0–6,000 lb. machine was used. A maximum loading before failure of 2,608 lb. on four sq. in., or 652 lb. per sq. in. was obtained. Deflection at failure was 0.023 in. Tests were run on ten specimens.

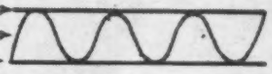
Impact strength was determined on a Tinius Olson weight drop tester, using a 20 lb. weight dropped on a 2 in. square anvil placed so as to cover an equal portion of fin and core elements. A new sample was used for each drop of the weight. Failure was obtained at 5 ft. lb. per sq. in. Ten specimens were tested.

(Continued on next page)

Typical Materials Used in Panel Construction

SAMPLE A

Plies — 2 fabric
3 fabric
2 fabric



Pressure — 325 lb. per sq. in.
Cure — 10 min. at 300 deg. F.
Weight — 16.1 oz. per sq. ft.

SAMPLE B

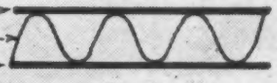
Plies — 1 aluminum — 1/16 in.
10 crepe paper
10 crepe paper
1 fabric

Aluminum sheet etched in 20 percent NaOH, then one coat brushed Bakelite resin 16320, dried 2 min. at 240 deg. F.

Pressure — 164 lb. per sq. in.
Cure — 10 min. at 300 deg. F.
Weight — 24.8 oz. per sq. ft.

SAMPLE C

Plies — 1 aluminum — 1/16 in.
10 crepe paper
1 aluminum — 1/16 in.

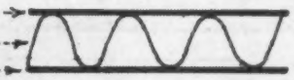


Aluminum etched in 20 percent NaOH, one coat brushed Bakelite resin No. 16320, dried 2 min. at 240 deg. F.

Pressure — 274 lb. per sq. in.
Cure — 10 min. at 300 deg. F.
Weight — 32.2 oz. per sq. ft.

SAMPLE C

Plies — 1 18 ga. S.A.E. 1010 steel
10 crepe paper
1 18 ga. S.A.E. 1010 steel

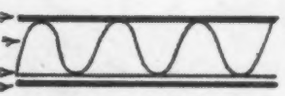


Steel brushed one side, one coat Bakelite resin No. 16320, dried 2 min. at 240 deg. F.

Pressure — 274 lb. per sq. in.
Cure — 10 min. at 300 deg. F.
Weight — 46.0 oz. per sq. ft.

SAMPLE D

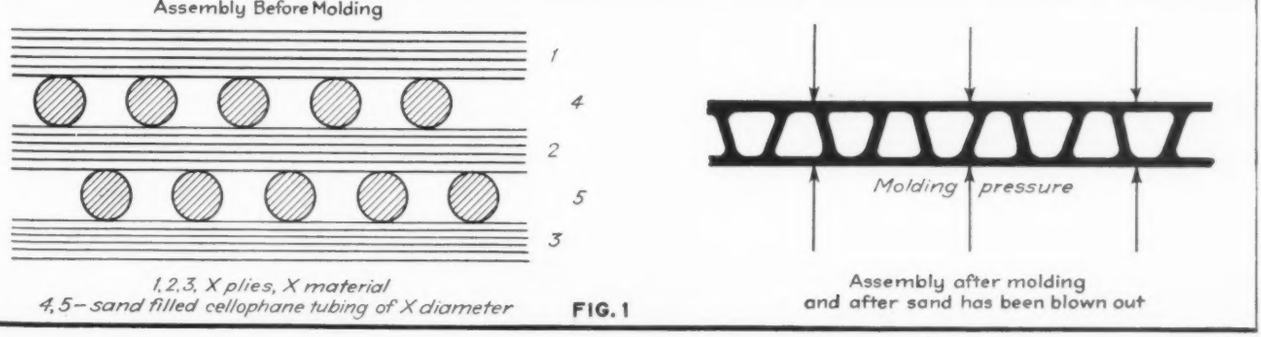
Plies — 1 steel — 18 ga. S.A.E. 1010
10 crepe paper
10 crepe paper
1 fabric



Steel coated one side Bakelite resin 16320 cut with 1-3 Solox, dried 2 min. at 240 deg. F.

Pressure — 164 lb. per sq. in.
Cure — 10 min. at 300 deg. F.
Weight — 34.8 oz. per sq. ft.

Fig. 1—Five-element sandwich. Each adjacent core is placed immediately over the void in the row below. Cores are loaded by use of a special fixture. Molding pressure forces plies in layer 2 into contact with plies in layer 3 and plies in layer 1, producing a celluar structure of approximately trapezoidal cross-section.



1, 2, 3, X plies, X material
4, 5 – sand filled cellophane tubing of X diameter

FIG. 1

Assembly after molding and after sand has been blown out

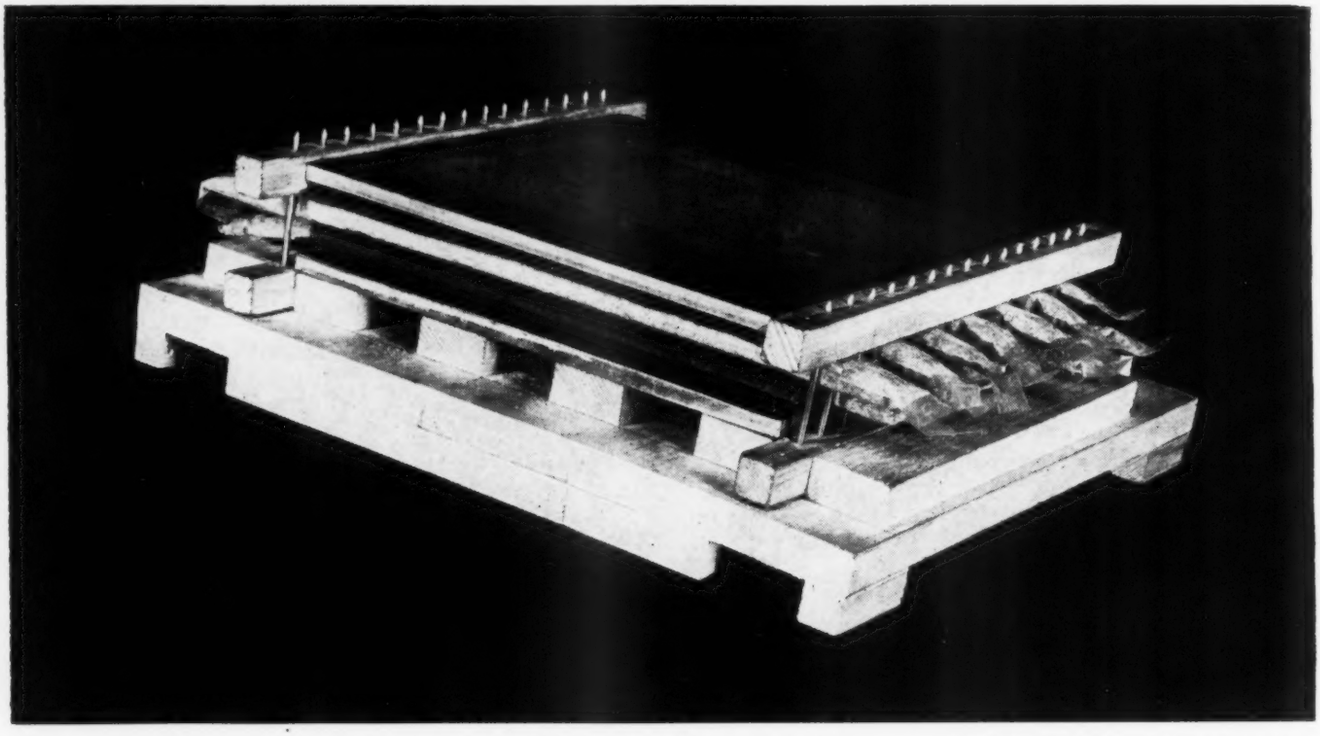


Fig. 2.—Assembly ready for molding. Top and bottom press pans are stainless steel. Positioning pins locate the sand-filled tubes. The entire positioning fixture is removed after approximately one-half the maximum compression is obtained, since there is no further slippage of tubing, and also to prevent squeeze-out of resin.

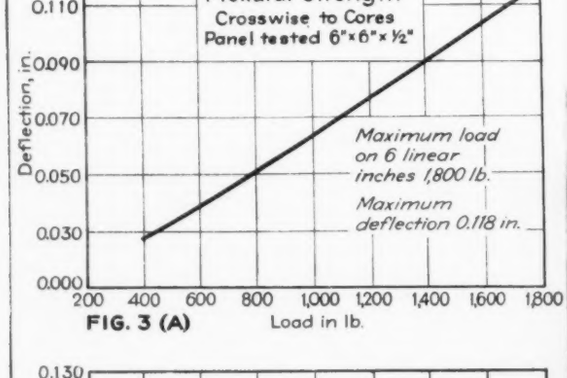


FIG. 3 (A)

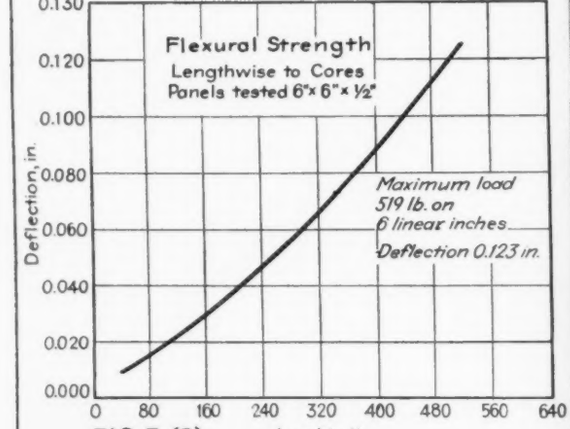


FIG. 3 (B)

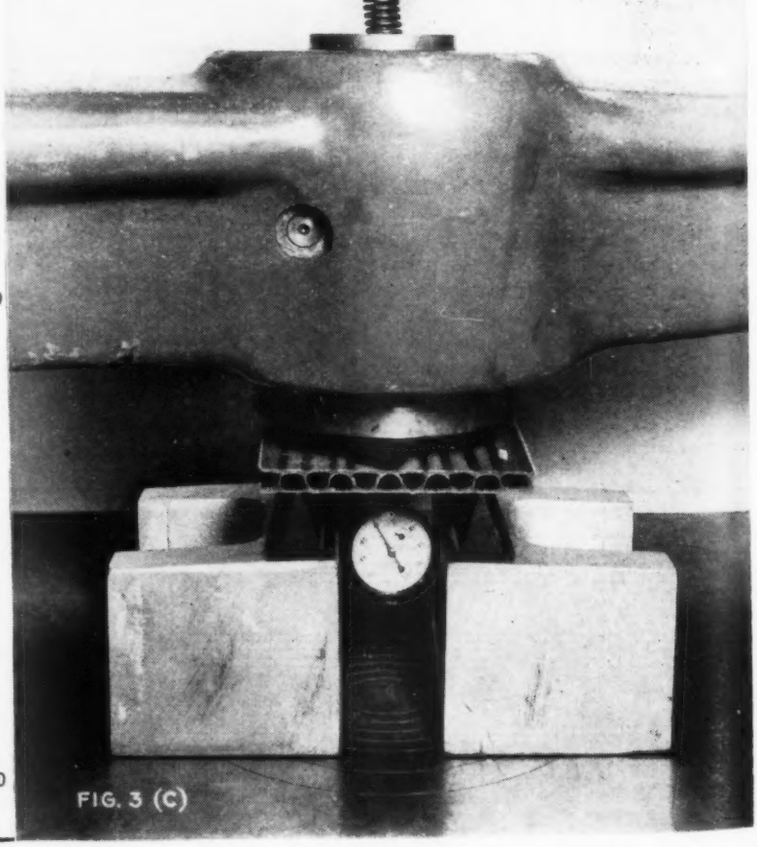


FIG. 3 (C)

Fig. 3—Flexural Strength. Contact surfaces on the V-blocks had 1/8 radius lower blocks were 4 in. apart. A Baldwin-Southwark Emery Universal testing machine was used. Upper V-block was placed substantially above the apex of one of the air cores. The lower V-blocks were placed so that the apex was immediately below the base of one of the air cores. Loading speeding 0.025 in. per min. Strength was determined in both directions of air-coring, on ten specimens.

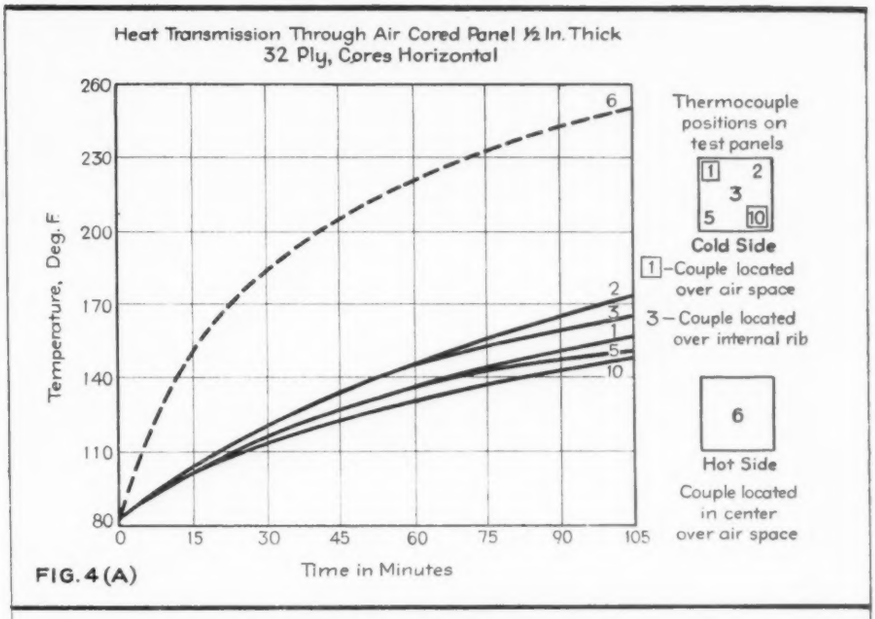


FIG. 4 (A)

Fig. 4.—Heat Transmission. Completely insulated boxes with internal dimension 6 x 6 x 20 in. were used. On one side of the box a 60 watt incandescent electric lamp was placed. In the center of the box the sample under test was located. One inch from the lamp, a baffle prevented direct heat transmission to the sample. Three samples were tested, each 6 x 6 x 1/2 in. One sample was tested with the air cores in a vertical position. The second sample was tested with air cores in a horizontal position and the third sample tested was solid. In the instance of specimens one and two, which contained the air cores, these

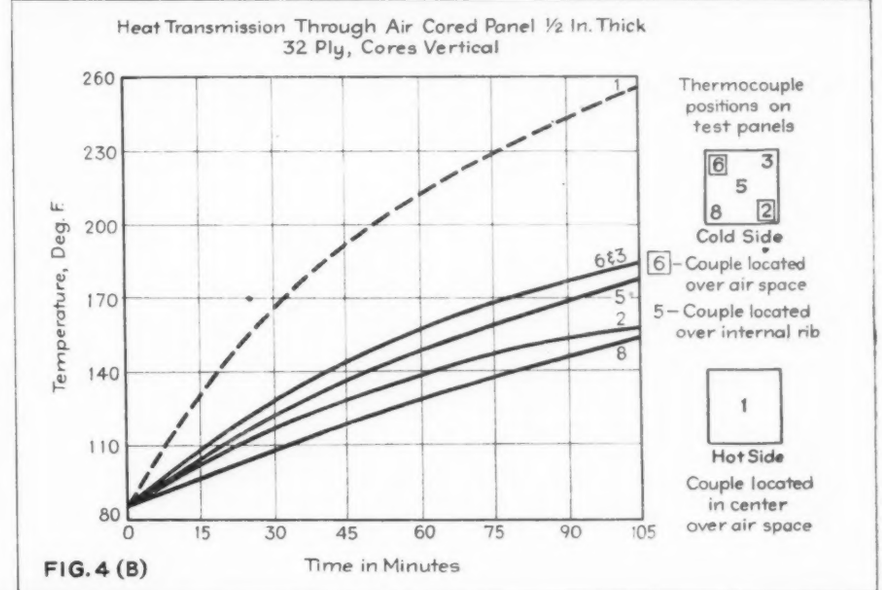


FIG. 4 (B)

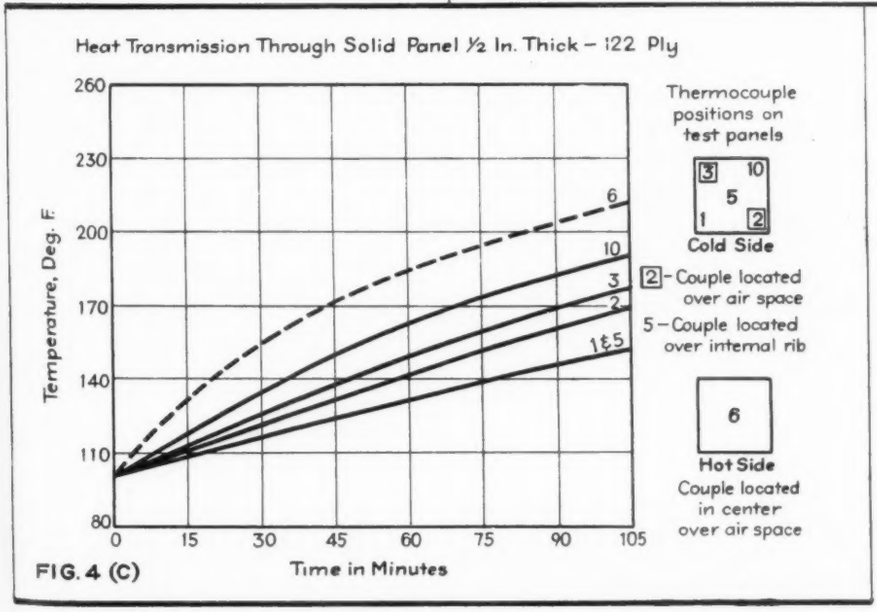


FIG. 4 (C)

were each molded by using 32 plies of resin impregnated material. The two surface sheets consisted of one external ply of fabric and then 10 ply paper. The interior of corrugated element consisted of 10 ply paper. The solid sheet was made from 120 plies of paper and one surface sheet of fabric on each surface. Six thermocouples were attached to the test specimen, using an especially prepared phenolic cement. The positions of these thermocouples are shown: 5 on the cold side and a single one on the hot side.

Dual Dimensioning of Drawings For Machining and Casting Uses

G. N. COLE
Chief of Design Operations, Pratt & Whitney Aircraft

Comparison of two general methods of dimensioning parts on drawings. A detailed description is given of the system in which three classes of drawing dimensions are identified. Casting tolerances and tolerances on concentricity are shown as a guide to the designer.

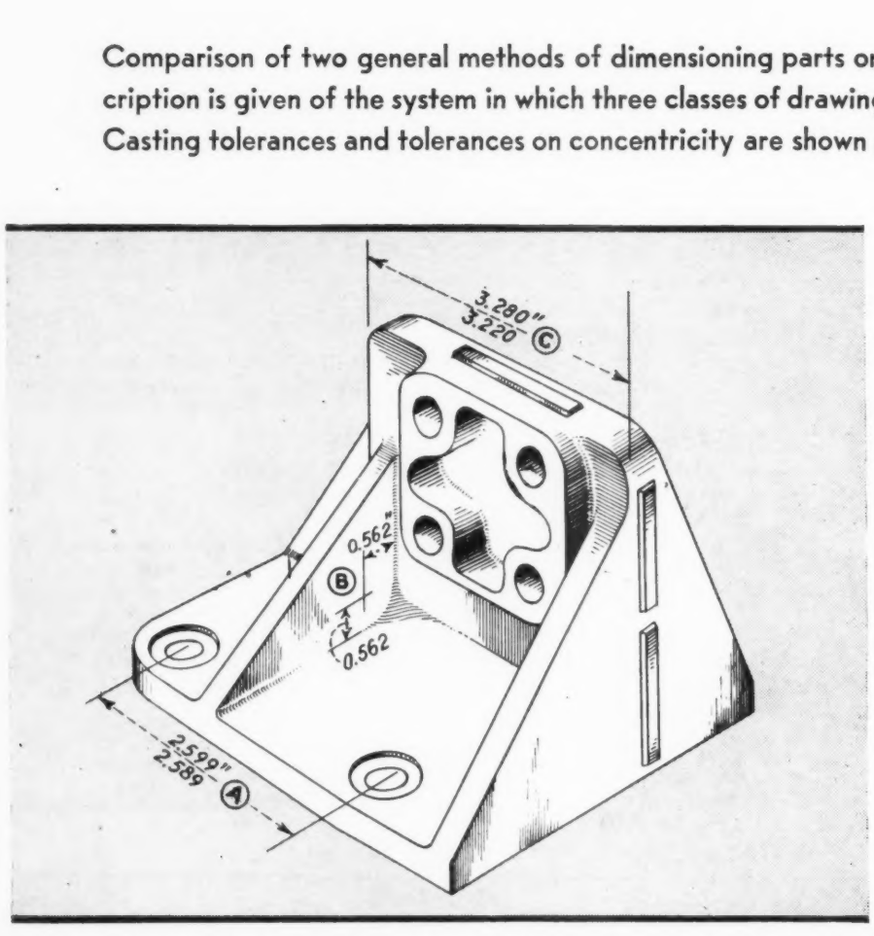


Fig. 1—A drawing illustrating three classes of dimensions at *A*, *B* and *C*. Class *A* specifies two limits and is a machining dimension. Whenever a class *A* dimension is needed in pattern construction, the mean value is used. Class *B*, a single-valued dimension, is used for pattern construction only. Class *C* defines surfaces as cast and has two limits. Note spread between Class *A* and *C* limits.

TWO GENERAL TYPES of decimal systems of dimensioning drawings are in use at the present time. One is the two-place decimal system in which the inch is graduated in tenths and hundredths; the other is the system in which the decimal dimensions are the equivalent of fractions, the smallest division being the sixty-fourth of an inch.

Pattern makers, die sinkers and other shop men carry scales graduated in sixty-fourths more frequently than scales graduated in tenths and hundredths. This is an argument often advanced to support the use of the latter system.

Whichever system is used, decimal dimensions to three places of decimals are available and advisable. Four places of decimals can be specified where necessary in precision design. Good drafting practice expresses each drawing dimension by two dimensions—a high and a low limit. A modification of this practice is to give the desired dimension and a tolerance, which in effect also provides high and low limits. In either case the designer or draftsman is required to note an allowable variation on each dimension.

An interesting outgrowth of this general system of dimensioning has been a system better suited to dimensioning combination casting and machining drawings. Although forgings are relatively simple when compared to most castings, usual practice has been to make separate forging and machining drawings. Casting dimensions, on the other hand, appear normally on the drawing used in machining the part. The dimensions for both casting and machining the part are designed in detail as they are drawn. Usually time does not permit making two drawings—one for the pattern maker and foundryman and the other for the machinist.

Basic General Notes

The requirements of the pattern maker, foundry man and machinist all differ, yet one drawing must convey specific information to all three. This is achieved through systematic dimensioning and three general notes that appear on each drawing. The notes, which form the backbone of the system, are as follows:

1. Pattern equipment must be designed to produce castings within the dimensions specified on this drawing.

2. The mean value of machining dimensions is a basic casting construction dimension where necessary.

3. Basic casting dimensions carry no tolerance.

These three general notes imply that the drawing will be read from two different points of view, first the pattern maker's and foundryman's, and second the machinist's.

MACHINING DIMENSIONS. Any dimension between a pair of machined surfaces such as finished surfaces and drilled holes is specified by two dimensions, which provide high and low limits. A dimension belonging in this class is identified by the letter A in Fig. 1. Fig. 2, 3, and 4 show other

typical machining dimensions, which are identified by the letter *A* also.

Pattern Dimensions. Dimensions used only in the construction of patterns carry no tolerance. They are termed basic casting dimensions although dimensions in addition to those having no tolerances generally are required to make the pattern. Wherever a machining dimension serves also as a pattern dimension, the mean value of the two limits is the pattern dimension. Since a mean value has no tolerance it can be considered as a basic casting dimension. Several basic casting dimensions are among the dimensions in Fig. 1 to 4. The letter *B* identifies some of them.

Casting Dimensions. Casting dimensions define surfaces as cast. They are two-valued like machining dimensions. In many cases they are located by dimensions used in machining the part or in the construction of its pattern. The mean value of them is used in pattern construction. Typical casting dimensions are illustrated in Fig. 1 to 4, in which the letter *C* has been used to identify some of them.

Casting dimensions generally have a spread between the high and low limits several times greater than the spread between limits on machining dimensions. On small simple castings tolerances less than ±0.030 in. sometimes can be held without difficulty. The following general casting tolerances have wide application and are offered as a guide to the designer in specifying the dimensions on drawings.

Castings up to 12": ± 0.030" tolerance
Between 12 and 15": ± 0.040" tolerance
Between 15 and 18": ± 0.050" tolerance
Over 18": ± 0.060" tolerance

In general, the tolerances depend upon the complexity of the casting. Wall thickness tolerances of ± 0.030 in. and core thickness tolerances of ± 0.030 are common on engine parts. This helps in maintaining alignment of cored openings in engine parts to within ± 0.030 in. Core tolerances based upon size of casting is standard practice with some companies. In establishing the tolerances on some dimensions of complex castings, an accumulation of tolerances may require the specification of larger tolerances, ± 0.060 or ± 0.090 in. usually being adequate.

Casting tolerances that define the limiting outlines of cast surfaces should be selected by the draftsman with care. Knowledge of the pattern design, expected core shifts, pattern wear, core abrasion from handling and foundry practices are of great help to the designer in establishing tolerances for casting contours. One pattern shop may design a pattern equipment that gives consistently one set of variations in castings, while pattern equipment of another design will produce the same part with different characteristics and variations. Changes should be made in the tolerances on the drawing to suit the circumstances.

A good time to make such changes is when sample castings are being checked at layout inspection. The draftsman can follow the inspection closely and then correct or modify the drawing as conditions warrant to satisfy pattern, casting, machining and design requirements. In some cases the design of the pattern equipment must be altered to get the desired product.

Dimension Reference Surfaces

Dimensions that define the casting on combination machining and casting drawings are referenced to appropriate target pads, the back of a flange, several bosses in the same plane, or a general wall. A fundamental in selecting the reference surface is that the surface must not be removed when the casting is machined. The first or basic machining cut is referenced to the same surface from which the other casting surfaces are dimensioned. The tolerance on this dimension is usually from ± 0.005 to ± 0.030 in., depending upon the prevailing conditions.

All other machining cuts are dimensioned either to the first machining cut or to some other machined surface. When complicated

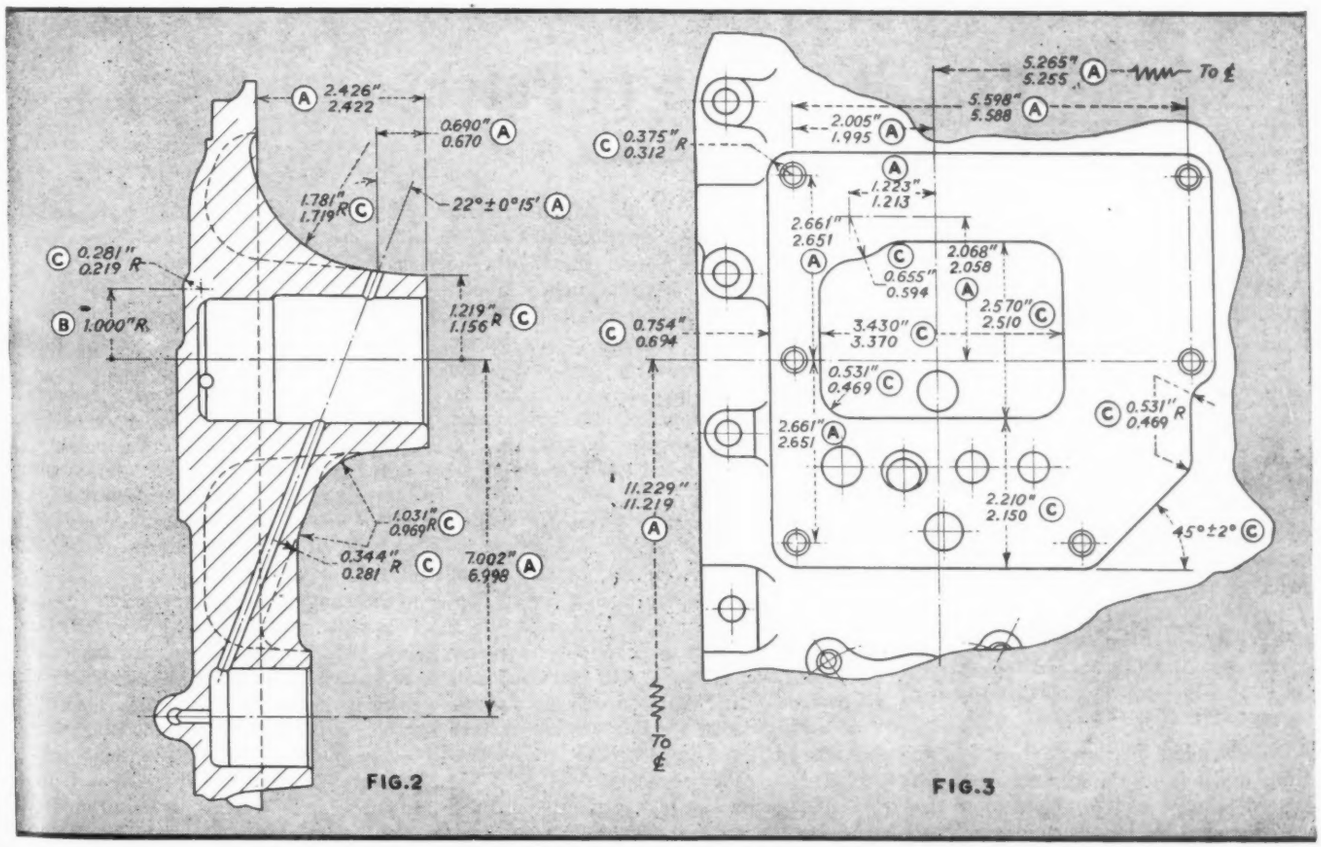


Fig. 2—A sectional view showing an angular tolerance for a class *A* dimension among the tolerances in all three classes. Fig. 3—A plan view in which the primary reference surfaces are outside the limits of the drawing.

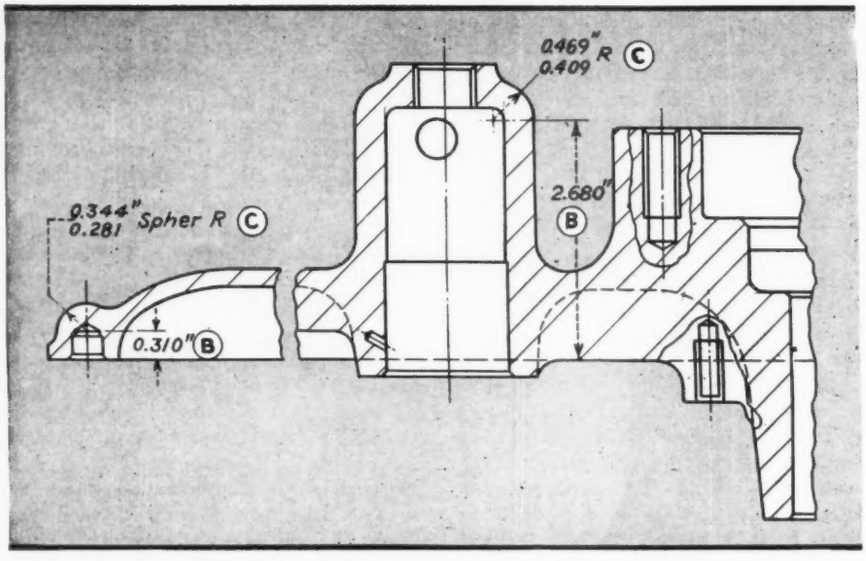


Fig. 4—A sectional view illustrating a *B* typical basic casting dimensions.

casting drawings are dimensioned, the use of secondary reference planes is permitted, providing the use of them is in keeping with pattern and foundry practices.

A note on the drawing, such as "start machining," often is advisable. It insures referencing the basic cut to the proper surface. It fixes a primary reference surface with respect to which any secondary reference surfaces can be established. It builds up cumulative tolerances in the same order that the designer used in establishing the limits for tolerances. It plainly identifies a machined surface that can be used in gage design or in parts inspection.

Such a note compels the draftsman to give thought to the machining operations and to the order in which they are performed. During his deliberations, he may discuss the matter with shop men, all of which is reflected in the design of parts that are dimensioned better and more readily manufactured.

Concentricity is important in the dimension of large castings, particularly when one surface is to be machined while another concentric to it is left as cast, when calculating tolerances to establish a minimum wall section required to carry a design load and when clearance is required between mating or adjacent parts. Under these conditions concentricity tolerances are applied. They are similar to the general casting tolerances and should be expressed as full indicator readings. The magnitude of the concentricity tolerance is usually twice the value for the corresponding general casting tolerance. Ordinarily it carries no sign.

Parts inspectors like drawings dimensioned in accordance with the foregoing system because the tolerances provide practical limits upon which to base acceptance or rejection of the parts. The tolerances supply information usable by tool and gage designers in making inspection gages and in devising the methods of inspection.

Case Histories In Patent Law

H. A. TOULMIN, Jr.

[EDITOR'S NOTE: These explanations of the law in specific patent cases are not intended to supplant the services of a patent attorney, whose accurate advice is necessary to handle individual patent problems. The purpose of these explanations is to give a better general understanding of patent law.]

NO. 1

QUESTION: Under what conditions are claims introduced into a patent invalid by reason of prior publication, use or sale more than one year prior to the amendment of the application and the introduction of the claims?

ANSWER: In the case of Muncie Gear Works, Inc. v. Outboard, Marine & Mfg. Co. et al, 315 U. S. 759, Mr. Justice Jackson stated the facts as follows.

The patent in suit was filed without the claims under controversy. They were later inserted by way of amendment more than two years after there was a prior public use and sale. The question was whether the claims added were thereby anticipated, or whether the date of invention of the claims could date back to the original date of filing.

In this case the patent as originally drafted contained claims that were limited either to a deflection plate or to an arched support which dealt with an anti-cavitation mechanism shown in the drawings of the patent. The drawings showed an outboard motor with an anti-cavitation plate located horizontally above the propeller. The cooling system had water passages enclosed in the housing but there was no cross-section of the housing.

The drawings showed an arched member extending from the housing and anti-cavitation plate over the top and to the rear of the propeller, This ended in a curved "deflection plate" extending rearwardly like a fixed rudder. The specification and claims described the action of this plate and said it prevented cavitation. But the specification did not assert that the plan was new or that it was employed in any novel cooperative relation to other elements.

Later, more than two years (at that time the Statute was two years but is one year now) after the prior use, the patentee's attorney inserted claims to the cavitation plate or the combination with the cavitation plate. The attempt was made to assert at that late time this was the essence of the invention.

Mr. Justice Jackson, speaking for the court, held the claims invalid because the original application "wholly failed to disclose the invention now asserted".

Under Section 4886 of the Revised Statutes of the United States, it is necessary that any new claim not so properly supported by the application, be inserted within one year form the date of prior publication. use or sale of the same subject matter by anybody. If the claim is a mere modification of existing claims, then it will date back to the filing of the application.

In Milcor Steel Co. v. Fuller, 316 U. S. 143, 147, the Court held that where subsequent claims changed the combination of elements in the claims originally filed, there would be a completely new invention set up.

The purpose of these decisions was to prevent the practice that has grown up of an applicant shifting his ground of invention after his application was filed when he once found that the prior art was against him as far as the claims were concerned that he originally put in the case.

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

Comparison of Roughness of Super-finished Plane Surfaces

Foreign Abstract, condensed from an article by J. Kluge and G. Bochmann in VDI Zeitschrift Bd. 88 Nr. 13/14, April 1, 1944, page 179.

SPECIAL SURFACES have been developed by the authors of this paper in connection with their investigation of the lubrication of machined surfaces. These surfaces have evenly distributed peaks and show a velvet-like, dull appearance as if they were sandblasted, although they are produced by a lapping process.

Customary roughness meters are usually not sensitive enough for measuring such surfaces where the height of the peaks is 40 micro-inches or less and—in addition—where the peaks are closely packed. Tracer methods fail because a tracer point with a radius of 400 micro-inches is too large. Interferometric methods can likewise not be used because interference strips cannot be obtained on dull surfaces.

A new instrument has therefore been designed and successfully used by the authors in their investigation of the effect of surface roughness on friction. This instrument does not give numerical values for the space height of the profile or for other data, but indicates comparative values which are sufficient for determining whether a machining method results always in the same surface quality.

It appeared that an instrument would be most suitable based on the principle of the "brilliance indicator", where a light beam is directed at 45 deg. on the surface to be investigated, and where the reflected light beam is picked up by an electric eye. The photo-current generated in the electric eye (and therefore the surface roughness) can be measured by means of a galvanometer.

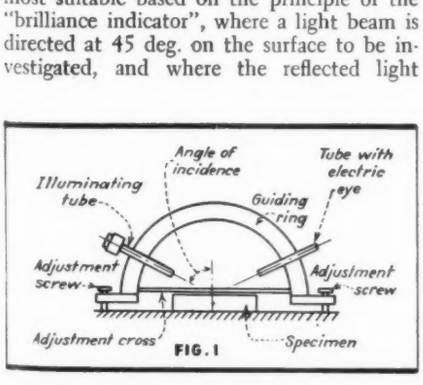


Fig. 1—Preliminary surface tester, employing reflection method.

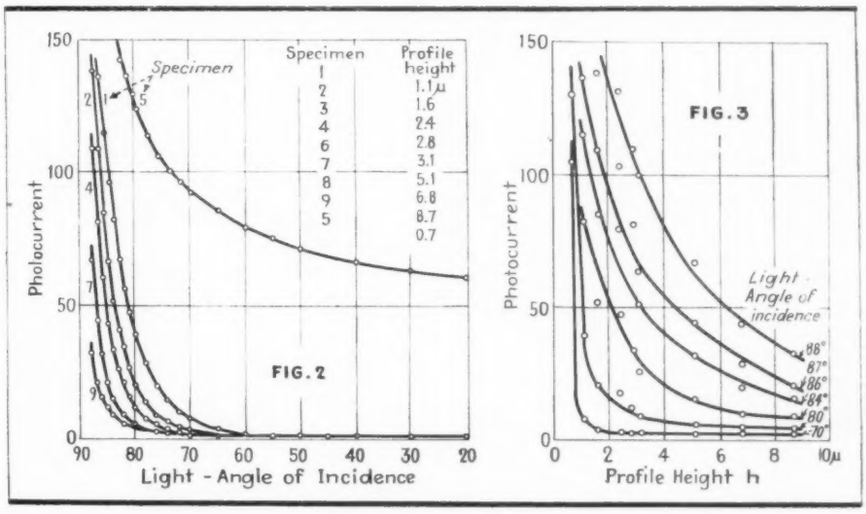


Fig. 2—Photocurrent registered by instrument (Fig. 1) as a function of angle of incidence for different specimens. Fig. 3—Photocurrent registered by instrument (Fig. 1) as a function of the height of the profile h measured with the Schmaltz surface tester and for various angles of incidence.

Since less light is reflected with increasing roughness, less current is generated. The current is proportional to the intensity of the reflected light.

However, such a "brilliance indicator" cannot be used directly in the case of dull surfaces, because of the lack of brightness of such surfaces. The sensitivity of the instrument would be much too low. The idea was conceived to use such an instrument with angles of incidence other than 45 deg., and an investigation was carried out for determining the angle giving maximum reflection on dull surfaces.

Fig. 1 shows the apparatus used for this preliminary investigation, where the angle of incidence (α) can be varied between 20 deg. and 90 deg. The adjustment cross is provided for adjusting the surface, which must lie in the axis of ring. The distance between surface and cross is measured with a gage and adjusted by means of the three adjustment screws on which the instrument is resting. The light bulb in tube is connected to a transformer which in turn is connected to the power line. A magnetic voltage controller is installed to eliminate voltage variations as high as ± 15 percent did not cause more than $\pm 1\frac{1}{2}$ percent variation at the galvanometer, while the variation amounted to ± 25 percent when the magnetic controller was disconnected.

Nine different cast iron surfaces were tested and the roughness measured with the ordinary method of oblique light. In addition, tests were run with the Schmaltz Instrument. It is difficult to measure surface roughness with the ordinary methods, as traces of the tool and fine cracks cannot be recognized. The height of the profile is likewise difficult to determine.

The most favorable angle of incidence was determined by plotting the electric (photo) current produced at the nine sur-

faces as a function of the angle. This is shown in Fig. 2. It is evident that angles of incidence less than 84 deg. did not give sufficient difference in current for clearly indicating the surface qualities; on the other hand, it was not possible to use angles greater than 88 deg. because the light beam was so much inclined in this case that insufficient light fell on the surface. A portion of the light passed directly into the electric eye without touching the surface.

In Fig. 3 the current is plotted as a function of the maximum height of the profile measured with the Schmaltz Instrument. Tracing along one of the curves of Fig. 3, it will be realized that a large deflection of the galvanometer needle indicates low roughness and vice versa. It must not be overlooked that a comparison of roughness in this way is possible only if the surfaces have been produced by basically the same machining processes.

An angle of 86 deg. was selected as the best angle for measuring superfinished, lusterless surfaces, having a roughness of about 40 to 320 micro-inches. At this angle the accuracy of the instrument is sufficient for measuring a range of 40 to 200 micro-inches. The galvanometer range is likewise satisfactory.

The final instrument consists of a horizontal ring of about 6 in. diameter. The illuminating tube and the receiver tube are both permanently fixed to the ring at the above mentioned angle. The instrument can be place on the workpiece and the distance between the surface and instrument can be so adjusted that the reflected light falls on the center of the electric eye. The amount of light and sensitivity of the galvanometer can be adjusted by changing the width of the slot in illuminating tube.

Nomographic Representation of Polynomials

From "the Nomographic Representation of Polynomials," by W. H. Burrows, in *The Journal of Engineering Education*, January, 1946, pages 361-75, published by the Society for the Promotion of Engineering Education.

THEORY AND APPLICATION of this method are presented in two parts. Part I describes a general method of constructing nomographs for polynominals in three and four variables. The construction of the nomograph for quadratic equations is as follows: (1) Two parallel linear scales are marked off at a distance, L, from each other, Fig. 1. (2) The distance, L, is subdivided into segments of length, s, measured from the left-hand scale such that s equals $Lx/(x+1)$ for the values of x within the range. (3) Straight lines are drawn through the origin of the left-hand scale to the points on the right-hand scale representing the values of $-x$. (4) The intersections of these lines with the vertical lines for corresponding values of x form a smooth curved scale for x. (5) The left-hand scale is used for values of B and the right-hand scale for values of A. A straight line connecting any value on the A-scale with any value on the B-scale will intersect the x-scale in a value of x which is a solution of the equation

$$x^2 + Ax + B = 0 \qquad (1)$$

The nomograph for the reduced cubic equation

$$x^3 + Ax + B = 0 \qquad (2)$$

is constructed in the same manner except that the ordinates of the x-scale are determined by the intersections of the vertical lines drawn from the origin of the B-scale to points $-x^2$ on the A-scale.

The nomograph for any polynominal of the type,

$$f_1(x) + Af_2(x) + B = 0 \qquad (3)$$

where $f_1(x)$ and $f_2(x)$ are any powers or other functions of (x), is constructed as follows: (1) The A and B-scales are laid off as before. The segments are $s = f_2(x)L/(f_2(x) + 1)$. The intersecting lines are drawn from B equal zero to the points A equal $-f_1(x)/f_2(x)$ to form the x-scale.

If the equation is

$$x^3 + ax^2 + bx + c = 0 \qquad (4)$$

a function of x may be defined by $f_1(x)$

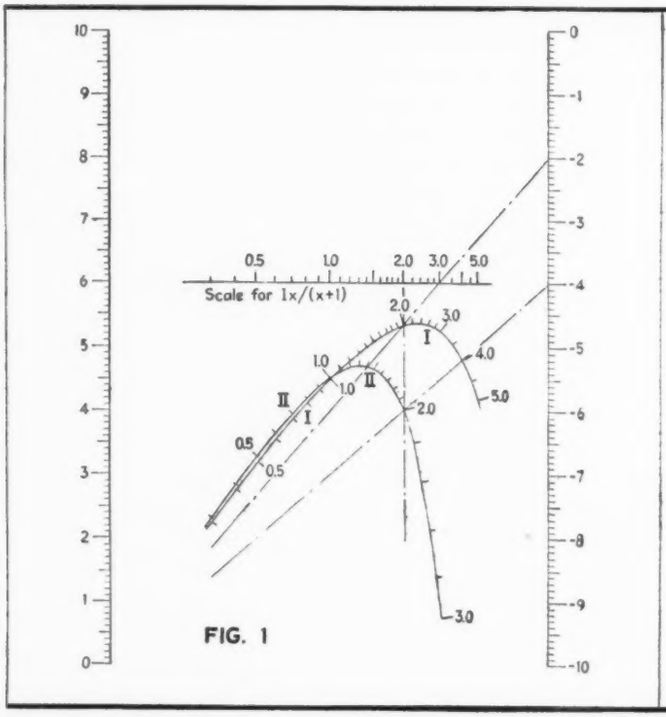


FIG. 1

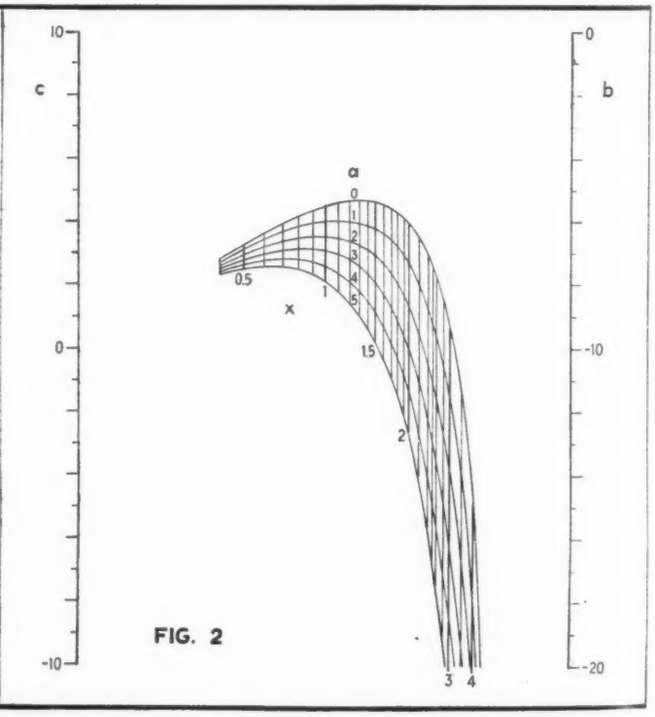


FIG. 2

Fig. 1—Method of constructing nomographs for the quadratic and reduced cubic equations. Fig. 2—Nomograph for the complete cubic equation, $x^3 + ax^2 + bx + c = 0$, illustrating the general method for polynomials in four variables.

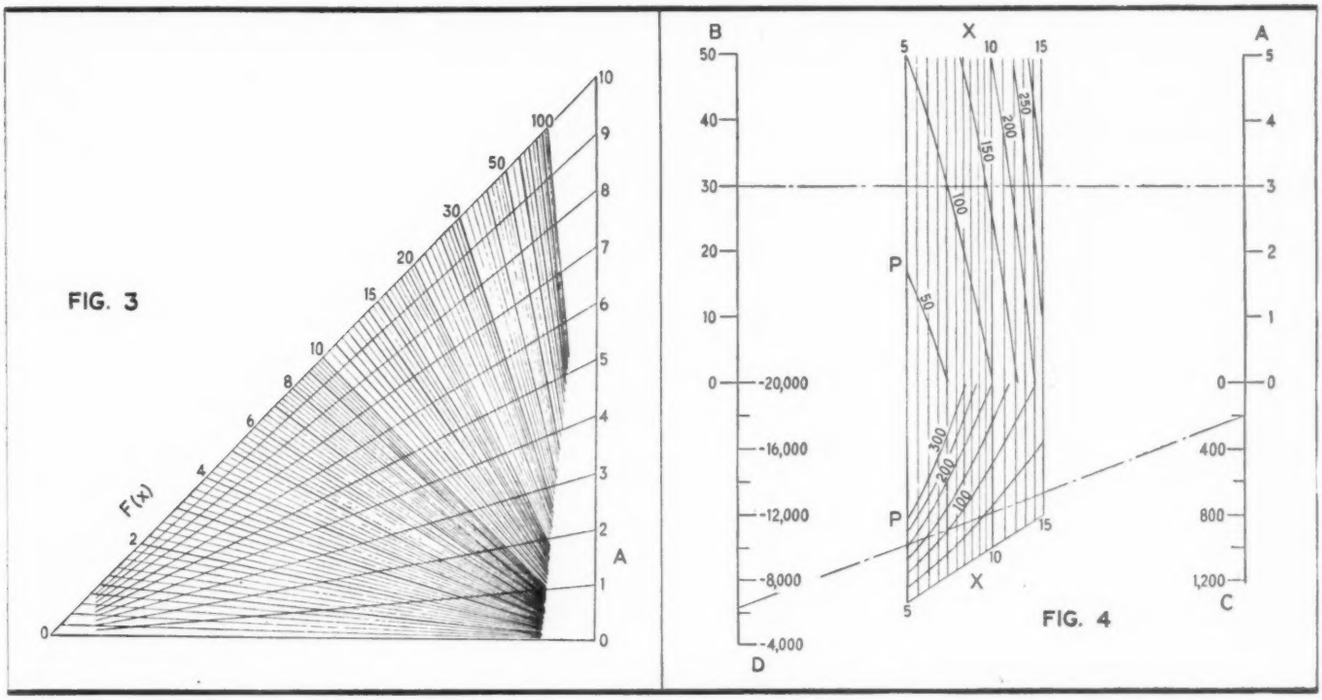


Fig. 3—The projected scale chart. Fig. 4—Second method for polynomials in five variables.

equals $x^3 + ax^2$, wherein the coefficient a is assigned a series of constant values, each giving rise to a separate curve for the x-scale. The resulting nomograph consists of two parallel extreme scales with a network scale between them, Fig. 2. The network scale has vertical lines for values of x, crossed by a family of curves of a values.

The segments are s equals $f_2(x)L/(f_2(x) + f_3(x))$ in the construction of the nomograph for the polynominal,

$$f_1(x) + Af_2(x) + Bf_3(x) = 0 \qquad (5)$$

The intersecting lines are laid through B equals zero and A equals $-f_1(x)/f_2(x)$ to determine the ordinates of the x-scale.

In setting up a nomograph for the equation

$$ax^3 + bx^2 + cx + q = 0 \qquad (6)$$

in which q is a constant term, the functions of x can be defined by $f_1(x)$ equal $ax^3 + q$, $f_2(x)$ equal x^3 and $f^3(x)$ equal x.

There is a relationship between the range of x and the ratio of the moduli of the A- and B-scales that gives an optimum distribution to the x scale for ease and accuracy of interpolation. A projected scale chart, Fig. 3, advantageously selects the optimum ratio of the moduli of the A and B-scales and also locates the abscissas of the x-scale without further construction.

For instance, to set up a nomograph wherein the range is to be 25 to 250, both scales would be multiplied by 10. Thus 25 would take the position 2.5 and 250 the position 25 in Fig. 3. The range would fall approximately in the middle of the line O to A equal 10, which line would be used in establishing abscissas of the x-scale. Then, since the values of both scales have been multiplied by 10, a ratio of 100 would be used in setting up the A and B-scales.

Part II treats two methods of constructing nomographs for polynominals in five variables. The sixth variable P is introduced and the polynominal is factored into two expressions, each containing x and P and two coefficients. The nomograph is arranged in two parts, on either side of a common scale, so that simultaneous solution will eliminate the parameter P. The two parts are set up in the manner described for polynominals of four variables.

In the second of the two methods the general quartic equation

$$x^4 + ax^3 + bx^2 + cx + d = 0 \qquad (7)$$

if factored into the parametric form:

$$(x^2 - P) + ax + b = 0 \text{ and}$$
$$Px^2 + cx + d = 0 \qquad (8)$$

where one part of the nomograph has the specifications

$$f_1(x) = x^2 - P,\ A = a \text{ and } B = b \qquad (9)$$

and the second part of the nomograph has the specifications

$$f_1'(x) = Px^2,\ A' = e \text{ and } B' = d \qquad (10)$$

Since A and A' are coefficients of the same power of x in the parametric equations, the following relationship exists: If the moduli of the a, b, c and d scales are in the ratio $m_a/m_b = m_c/m_d$, and the c and d scales are colined with a and b scales, respectively, the abscissas of x for both parts of the nomograph will be identical.

Fig. 4 shows the nomograph, the scales and ratio of the moduli for which were determined by the projected scale chart, Fig. 3. To solve for x, straight-edges are laid from a to b and from c to d. The value of x at which the straight lines simultaneously intercept identical values of x and P is a root of the equation.

Similitude and Experimentation

Foreign Abstract condensed from "Theory of Similitude as the Basis for Experimentation" by M. V. Kirpitchef, *Bulletin of the Academy of Sciences, Class of Technical Sciences,* (Russia) 1945, No. 4-5.

RESULTS OF AN EXPERIMENT dealing with phenomenon can be extended only to similar phenomena. Thus, the theory of similitude must be the basis for experimentation. The theory can be applied only when the parameters characterising the phenomenon can be related by a mathematical equation.

Three theorems of similtude can be applied in the experimental study of phenomena. The first two indicate how the experiment is to be set up and how the results must be analyzed. The first theorem concerns the existence of similtude; the second concerns the existence of the so called criterial equation. The experiment must establish all the magnitudes constituting the criteria of similitude, and the results of the experiment must be expressed in the form of a criterial equation. This being done, the data furnished by the experiment can be extended to all phenomena analogous in nature.

The foregoing conclusion will remain purely formal until a means of recognizing the similitude of the phenomena can be found. It is true that similitude can be

established by means of experiment, but this is tantamount to repeating the experiment from the beginning. It is obviously of the greatest consequence to be able to recognize similitude on the basis of minimal data.

The third theorem establishes the conditions necessary and sufficient for the existence of similitude. It is distinct from the former two, which are based on the general equations. Since the establishment of similitude requires the comparison of concrete phenomena, each of them must be isolated from the multitude of phenomena described by the general equations. This can be done by adding to the general equations the conditions of mono-valuedness related to the phenomenon selected.

For similitude of phenomena the magnitudes in these conditions of mono-valuedness must obviously be similar, but this is not sufficient. Similar phenomena have numerically equal criteria of similitude. Among these criteria there may be such as are made up solely of magnitudes in the conditions of mono-valuedness. These criteria must be then identical. The third theorem establishes the sufficiency of these conditions for similitude between the phenomena under consideration.

As applied to machines the third theorem is embodied in the low of model and furnishes a means of establishing similitude between models and nature. Obtaining conditions of mono-valuedness is a problem of theoretical physics. The differential equation can be integrated only when mono-valuedness conditions are established. This means that the starting point for the theoretician and the experimenter is the same.

Radio Relaying

From "The Outlook for Radio Relaying" by Ralph Brown, Director of Radio and Television Research of Bell Telephone Laboratories, in October 1945 issue of *Bell Laboratories Record*, 463 West St., New York 14, N. Y.

PERMITS for microwave radio-relay system between New York and Boston is practical attempt to apply to radio the repeatered-circuit methods for wire transmission and universally used in long-distance telephony. The significance of the undertaking is interpreted by Table I, giving the sweep of radio advances.

The microwave radio relay may find its best place near the middle of other types of repeatered transmission systems. Estimated characteristics of an assumed microwave circuit being 4,000 mc. transmission with antenna apertures of 100 sq. ft. show that the radio-relay system would have 25-30 mile spacing between repeaters, a 2½-3 db. per mile attenuation, and about a 70 db. repeater gain. The repeater spacing and average attenuation of this hypothetical radio-relay system falls between the voice frequency and carrier circuits used on toll cables. However, its band width is more like that of the coaxial cable. Actual characteristics in practice and system cost are still uncertain. Also, until low-distortion amplifiers are available for microwaves, multiplexing will have to use other than usual carrier methods.

Table I

Trend of Radio Performance Characteristics

		1915	Early 1920's	Late 1920's	1930's	1940's
		Trans-atlantic Radio-Telephone Demon-stration	Ship-Shore Experiments Broad-casting	Trans-oceanic Service	F.M. Television Virginia Capes Multiplex	Radio Relays and ?
	Ideal Require-ments for Wire Type Performance	5,000 Meters (60 kc.)	500–200 Meters (.6–1.5 mc.)	60–15 Meters (5–20 mc.)	7–2 Meters (40–150 mc.)	15–3 Centimeters (2,000–12,000 mc.)
Directivity	Sharp	None	None	Some	Moderate	Sharp
Noise	Low and Steady	Severe Static Variable	High Static Variable	Moderate Static Man-Made Interference Important	Low Static Man-Made Interference Serious	None (As Yet)
Transmission Disturbances	Small and Slow	Small	Marked Rapid	Severe Rapid	Echoes Troublesome for Television	Small and Slow
Frequency Space Available	Unlimited	Picayune	Meagre	Small	Moderate (Crowded)	Large

Low-Pressure Molded Laminates

From "Present Products in the Low-Pressure Molded Laminates" by W. J. Connelly, Bakelite Corp. Presented Feb. 2, 1946 at the Low Pressure Conference, Society of the Plastics Industry.

THREE GENERAL TYPES of resins are suitable for low-pressure laminate molding:

1. Phenol-formaldehyde.
2. Amines: a. Urea; b. Thio-urea; c. Melamine-formaldehyde.
3. Thermosetting ester resins.

Depending on the resin used, pressures needed for proper lamination and successful production may range from ¼ to 500 lb. per sq. in. Two factors determine the resin's suitability for low-pressure laminating:

1. High flow at relatively low volatile content and pressure.
2. Maximum bonding strength at a given low molding pressure.

The strength and toughness of molded laminates are proportional to both the resin and the filler. In general, excess acidity of the resin should be avoided because of its tendency to increase the brittleness of cellulose fibers. Some thermosetting resins, when used in glass cloth laminates, show excellent strength characteristics, while the same resins may cause embrittlement of cloth or paper base laminates.

The sizing of fabric definitely affects the strength of laminates. When a fabric is used with phenol formaldehyde resin, tensile strength of the unsized laminate is 25 per cent higher and impact strength is 30 per cent lower than the sized-fabric laminate. It is possible to increase toughness by sizing fabrics with thermoplastic resins and bonding with thermosetting resins.

Alkaline paper speeds up the cure of some resins. Acid paper is to be avoided because within certain limits it may alter the cure of some resins, to cause delamination or poor strength. It is more practical to use neutral paper and obtain a desired pH by adjusting the resin.

The bulk factor or density of the filler influences, to some extent, the minimum pressure that may be employed. In general, increased bulk means increased resin content to effect flow at any given low molding pressure. Low resin, high density papers or tightly woven fabrics mold at lower pressures than the waterleaf type of paper or loosely woven fabrics, although materials having a low density are likely to spread and fill out the contours of the mold better. This is especially true when molding complicated shapes.

Fabrics and matting used in low-pressure molded laminates depend on specific need for specialized properties, such as electrical and mechanical properties.

Five methods are used in low-pressure laminating:

The autoclave method utilizes a female or male mold and rubber bag. Such items as antenna mast housings are molded of glass cloth in sheet metal welded molds using this method.

Second, the vacuum pressure method in

which a male mold is used with a rubber bag. A partial vacuum is created between the bag and the mold, producing an effective pressure on the outside, up to 14.7 lb. per sq. in. (atmospheric), depending on how perfect a vacuum is obtained. Instead of an autoclave, a plain oven is used. This method is used in making boast by sandwich construction.

The third method is the "hat press" method in which molds, either male or female, are used with a vulcanized rubber bag in a toggle press. Instead of air within the rubber bag, hydraulic pressure is used. Open gas flame heated platens or steam channel platens are used, and most molding is done with female molds, although both male and female molds, or two female molds are used.

The fourth process is the conventional multiple opening or single opening hydraulic press in which plain flat surfaces are produced. This type of equipment has often been passed over when considering low-pressure molded laminates but it is extremely important. Currently there are presses in operation with platen working surfaces 8 feet by 18 feet.

The fifth process is the continuous lamination method. Here the separate layers of material pass through treating tanks, merge at squeeze rolls with a top and bottom layer of cellophane, which is always wider and longer than the material being laminated. Leaving the squeeze rolls, the assembly travels through a heating tunnel.

A sixth process, while not truly low-pressure laminating, should be included. This is post-forming. Using simple wooden, laminated or light metal molds, the residual thermoplasticity of high-pressure laminated flat stock may be utilized by external heating of the sheet, and forming while heated.

Friction Welding of Plastics

From "Friction Welding" by Robert N. Freres, Rochester Button Company. Presented Feb. 7, 1946 at the Canadian Section meeting, Society of the Plastics Industry.

To WELD two thermoplastic parts together, the surface temperature of each piece must be raised to its molding temperature, pressure applied and the parts cooled. In welding, this surface heat can be applied by friction. For example, rotating a plastic rod in a drill press at high speed against a stationary part will develop frictional heat and raise the surface temperature to 300 deg. F. in a few seconds. If both parts are then brought to rest and pressure applied, they can be welded.

The heat generated by friction is the product of the coefficient of friction, the pressure and the distance moved. The time factor is not included. To reach a high temperature this friction must be applied at a rapid rate, so that the rate of heat input exceeds the heat dissipated.

Frictional heat represents an improvement over other types of surface heating. The heat is generated so rapidly that very little heat is dissipated during cooling, resulting in a rapid production operation. Also, only the immediate surface is heated. A delicately molded part of thin section will not distort, since the heat does not penetrate deeply. The overflow or flash will be thin and easily broken off, making for an easy finishing operation.

A third advantage: As the material is heated and reaches molding temperature it begins to flow under the light frictional pressure. Thus any dirt or oil present on the surface is removed, exposing a clean surface for the final weld. This is especially valuable when welding transparent plastic.

Fourth, the high temperature obtained does not cause chemical changes, because air is excluded from the contacting surfaces. Nylon, for example, which is sensitive to oxygen at high temperatures, is easily welded at temperatures up to 500 deg. F.

Cellulose nitrate and other flammable plastics can be welded submerged in water. The quality of the weld does not suffer. Liquid filled plastic compass cases have been welded submerged in the proper liquid. The part is removed as one solid homogeneous piece completely filled with liquid, containing no air bubbles. Plastics that weld at 40 deg. F. have been welded in gasoline and yet do not burn because there is no air for combustion at the hot surface.

Tests show that methyl methacrylate welded by friction is practically as strong in tension as the solid rod. The impact strength is slightly lower than the solid rod but twice as good as a cemented joint. Light transmission equals the cast rod. All thermoplastic materials of the same chemical composition can be welded by frictional heat and pressure.

A further advantage of frictional heated welding is that rough surfaces can be used. If oversize parts are welded and the heating prolonged, the excess material can be removed during the welding.

If one of the parts to be welded is circular, it should be rotated in a drill press or lathe against the stationary part. A rubbing velocity of about 1000 ft. per min. should be used. When the plastic begins to overflow, power should be cut off and about 300 lb. per sq. in. pressure applied. A few seconds after the parts come to rest the weld is completed.

Any shaped piece can be welded if oscillation is possible. Electrical vibrators of the proper size are suitable for welding irregular parts. An eccentric attached to a drill press has also been used. Non-uniform rubbing velocity over the entire area is compensated for, because the area receiving the greater rubbing heats rapidly and flows away, increasing the pressure on the cooler area, which in turn generates more heat. The result is a uniform surface temperature, despite the unequal rubbing velocity.

A thermoplastic material can be softened by rubbing against a solid non-softening part; upon applying pressure a bond is obtained. As in welding, the flow of heated surface material is required to bond thermoplastic materials to non-softening materials. A plastic knob can be bonded to a wooden handle by rotating the knob against the wood until the plastic flows and then applying pressure.

It has been found that the hot liquid plastic will penetrate deeply into wood or porous material. A plastic rod bonded to porous bronze penetrates to a depth of one-eighth inch. Following this procedure thermoplastics can be bonded to materials such as wood, glass, metals and ceramics. Brass tubing can be fed directly into the plastic while rotating at high speed. The tube becomes bonded on both inside and outside surfaces and is difficult to remove. The special advantage of friction is that the tube cuts its own path into the plastic, displacing the material as it goes. Thus the plastic surrounding the insert is not as highly stressed as it would be if a hot tube was pressed into the plastic.

Need For Engineers

From "Do We Really Need Engineers and Scientists" by Allan R. Cullimore, in *The Journal of Engineering Education*, January 1946, pages 320-26, published by the Society for the Promotion of Engineering Education.

THE SHORTAGE of engineers and scientists is so great that it will certainly affect not only the internal economy of the country, but will seriously impair the United States as a world power by limiting its capacity to produce and to supply world markets. The returning veteran offers perhaps the easiest solution to meeting the deficit, which amounted to 20,000 engineers last year.

By 1949, the earliest possible date for graduating sufficient engineers to meet the yearly demand, the minimum deficiency of 50,000 engineers is predicted. In many industries the feeling is that more engineers will be needed in research and development. In other industries engineers—technical men—will be needed, particularly in the lines of distribution, with special emphasis on the need in foreign countries and markets. The averages seem after this fashion: The smaller companies to find the greatest need for increased engineering personnel; the medium sized companies less and perhaps the larger companies the least. These estimates are spread over a wide range—from 200 percent in the smaller companies to 50 percent in some of the larger ones.

The capacity of the engineering colleges to train men, the effect of available jobs and wages upon the number of ex-service men that seek education, rejections because of educational deficiencies or because of deficiencies in aptitudes and attitudes, the martial status of ex-service men, all influence the college enrollment. Legislation, executive orders and public education all have their proponents as to the method of correcting what may be called a desperate situation.

Without even indicating that another war is possible, or that the dropping of another atomic bomb is within the limits of imagination, it is a fact that realistically this country must maintain its place in the march of scientific and technological progress.

Resistance Coefficients and Flow of Liquids in Pipe Systems

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Consulting Engineer

Method for calculating size of pipe lines to obtain a specified flow of liquid. Resistance coefficients are given for discharge orifices, pipe enlargements and contractions, diaphragms and perforated plates, strainers and common fittings. Examples are included to show how the data are applied in the solution of flow problems in pipe systems.

A SUBSTANTIAL PART of the general equations used in the solutions of problems dealing with the flow of fluids in pipe lines consists of friction and resistance coefficients that have been established by experiments.

In general, the mean theoretical flow of a fluid in a piping system or a part thereof can be expressed in terms of the actual mean velocity and a resistance factor. Thus with

v_t = mean theoretical velocity, ft. per sec.
v_a = actual mean velocity, ft. per sec.
μ = resistance factor

$$v_a = \mu v_t \qquad (1)$$

Since pipe lines consist of many different elements the resistance factor μ in Equation (1) must take into account the individual resistance coefficients of all the parts in the system. This can be done when

F_n = resistance coefficient of any part of the system designated by the subscript n.
ΣF_n = sum of the resistance coefficients of the parts considered.

and defining μ by

$$\mu = \sqrt{\frac{1}{\Sigma F_n}}$$

Values of resistance coefficients for different parts of a piping system are:

1. DISCHARGE ORIFICE. Values given include the effect of contraction of flow.

A. Free discharge from end of pipe

$F_{1a} = 1$

B. Free discharge through round orifice in thin bottom of tank

$F_{1b} = 1.69$

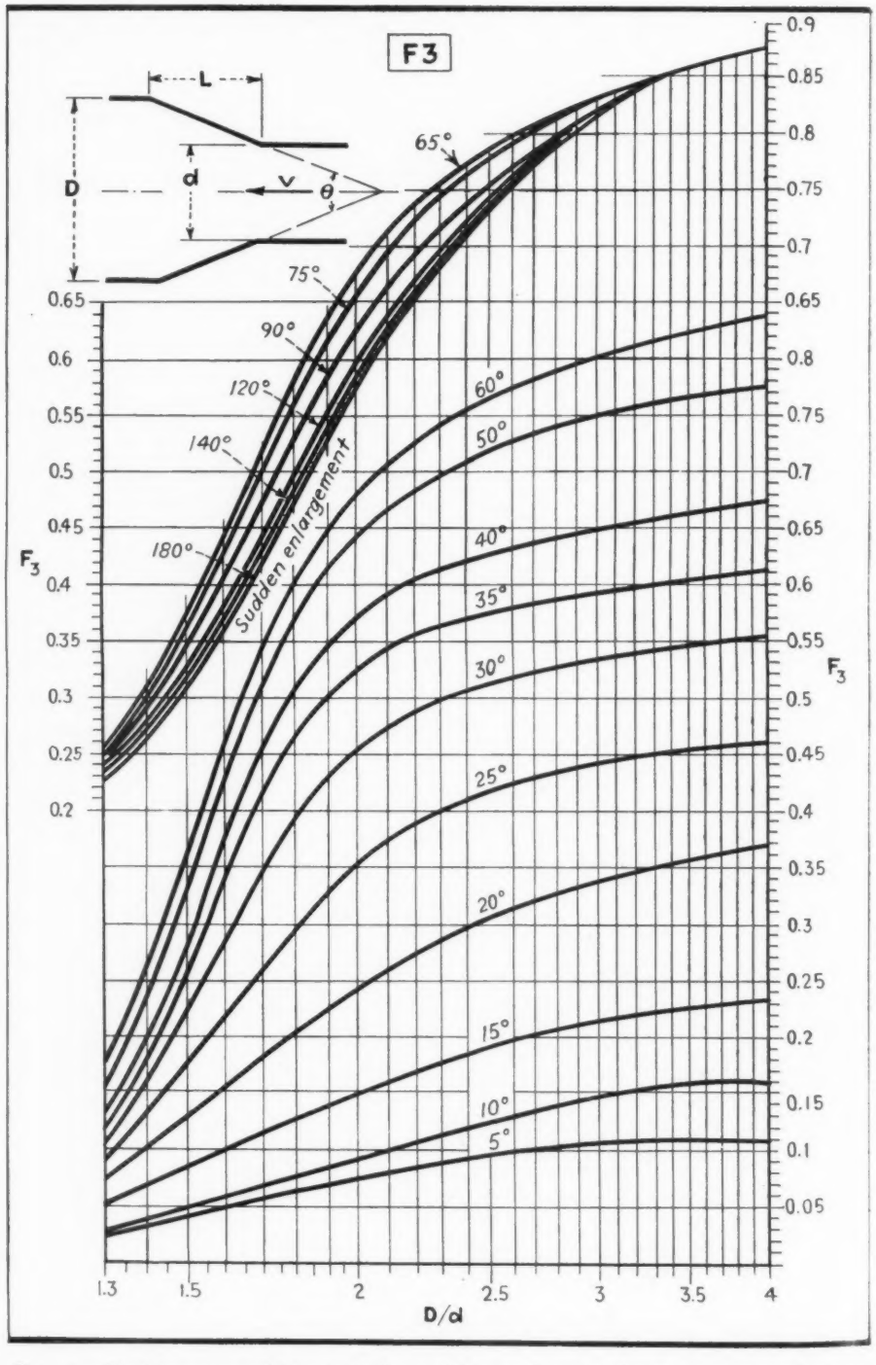


Fig. 1—Resistance coefficients F_3 for sudden and gradual enlargements of pipes. See Fig. 2 for angle θ when L, D, and d are known.

C. Free discharge through thin side wall of tank

$F_{1c} = 1.84$

2. Fluid Entering Pipe From Tank.

A. Pipe fixed to tank

$F_{2a} = 0.49$

B. Pipe fitted with thick flange in which the internal diameter d increases to 1.28d, and radius of internal curvature is 1.63d.

$F_{2b} = 0.06$

F_{2b} is calculated for the velocity of flow at the section where area equals $0.785d^2$.

3. Sudden and Gradual Enlargements of Pipes. Values of resistance coefficients F_3 for pipe enlargements are given in Fig. 1 for different ratios of large diameter D to small diameter d and angle θ of taper. When the ratio of length of taper L to the inside diameter d is known, Fig. 2 can be used to find θ. F_3 is calculated for the velocity of flow in the small section of the tube.

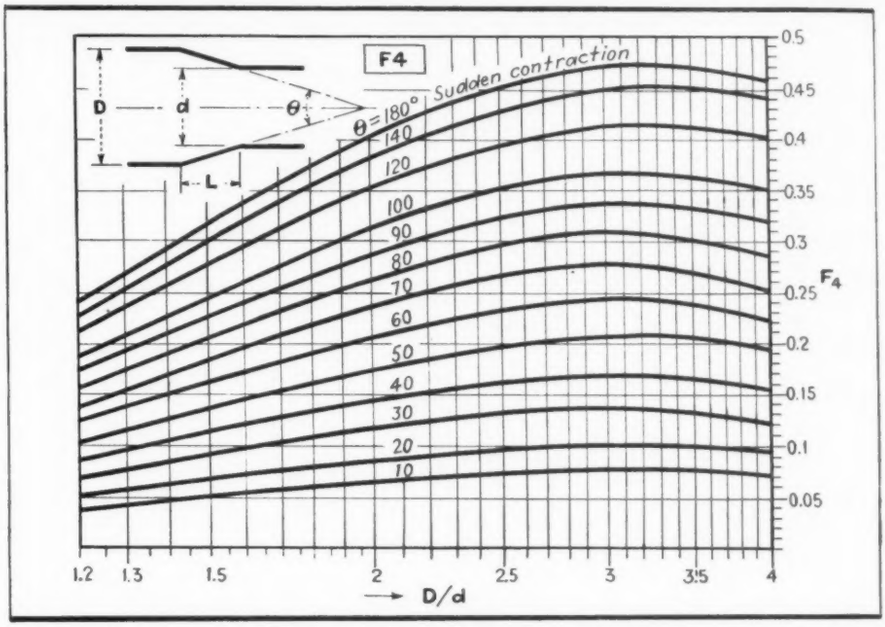


Fig. 3—Resistance coefficients F_4 for sudden and gradual contractions of pipes. See Fig. 2 for angle θ when L, D, and d are known.

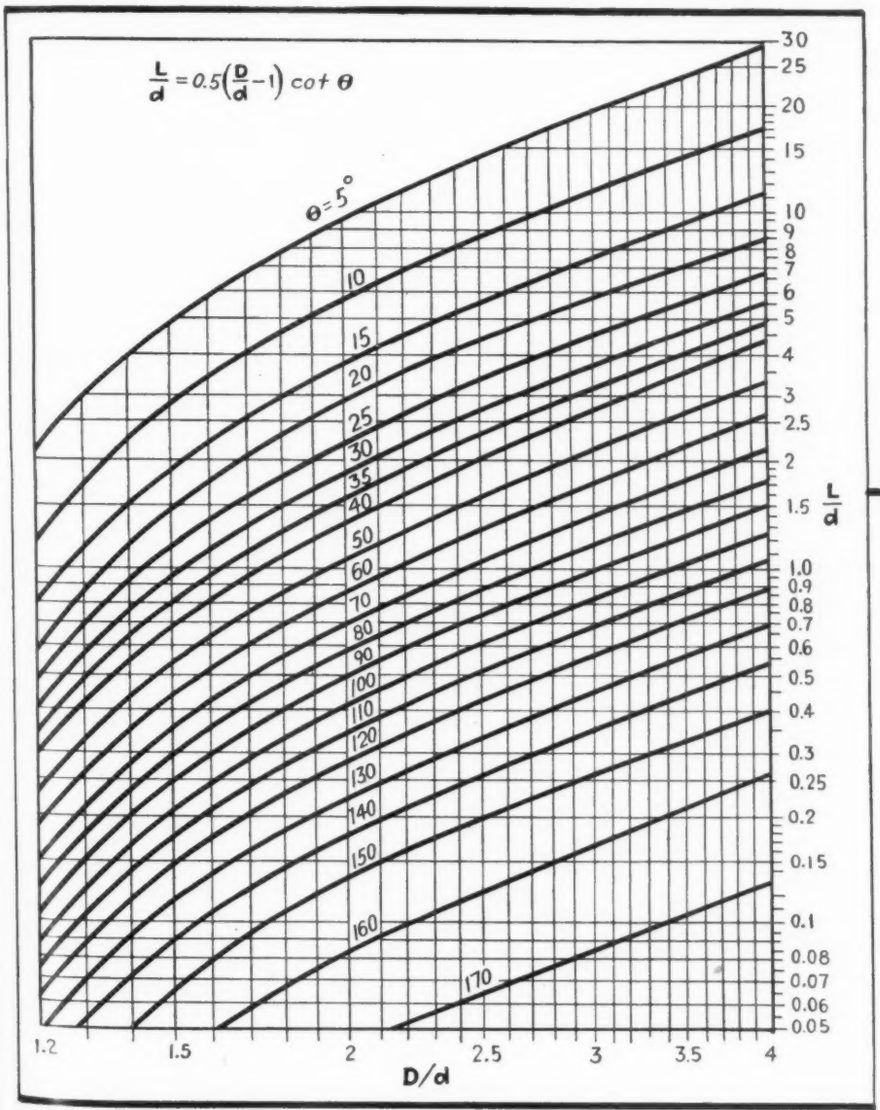


Fig. 2—Enlargement angles in terms of L, D, and d.

4. Sudden and Gradual Contractions of Pipes. Values of resistance coefficients F_4 for contractions in pipes are given in Fig. 3. F_3 is calculated for the velocity of flow in the small section of the tube.

5. Diaphragms and Perforated Plates in Pipes. Values of resistance coefficients F_5 for diaphragms of different thicknesses in pipes are given in Fig. 4 for different ratios of d^2/D^2, where d is the diameter of the hole

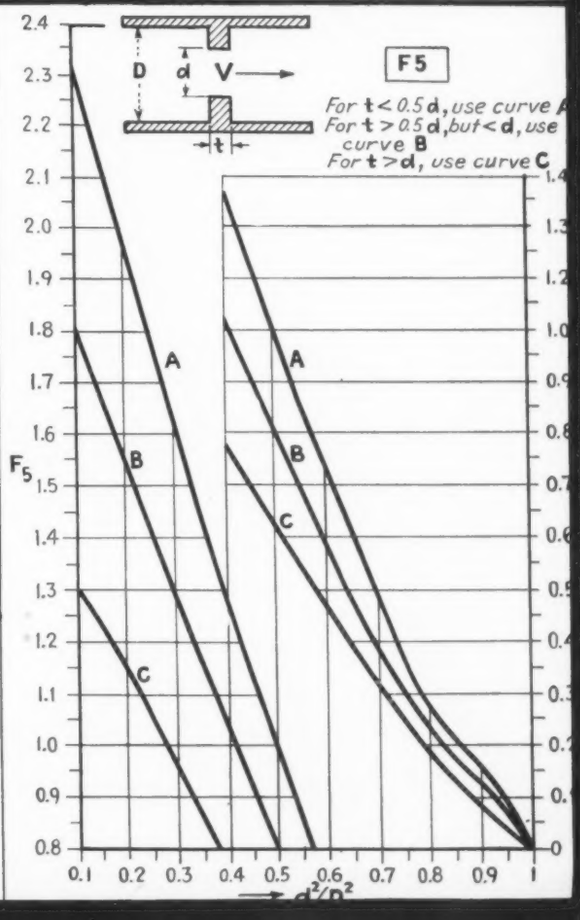


Fig. 4—Resistance coefficients F_5 for diaphragms in pipes.

in the diaphragm and D is the inside diameter of the pipe.

For diaphragms that have a number of holes instead of one central hole,

s = total passage area of holes
S = area of inside of pipe
$d^2/D^2 = s/S$

For example, if a wire gauze is fixed between pipe flanges, and the ratio of passage area in the screen to the area of the inside of the pipe is 0.68, and the thickness of the wire gauze is less than 0.5d, then where the ordinate for d^2/D^2 equals 0.68 intercepts curve A, F_6 is read as 0.54.

6. Fittings and Branches of Pipes. Values of resistance coefficients F_6 for fittings and branches of pipes are given in Fig. 5. Note the value of F_6 depends upon the direction of flow.

7. Fluid Friction in Pipes. In Fig. 6 values are given for a factor f for different velocities of flow, these values apply to new iron or steel pipes. Where

L = length of pipe, ft.
d = internal dia., in.

the resistance cofficient F_7 is found from

$F_7 = fL/d$

For copper or brass pipes, the values of f as given in Fig. 6 are multiplied by 0.85; for cast iron pipes multiply by 1.15; for old pipes multiply by 1.30. If oil is flowing in the pipes multiply values of f by 0.9.

Fig. 5—(Right) Resistance coefficients F_6 for pipe fittings and branches.

Fig. 6—(Below) Values of factor f for calculating resistance coefficients F_7 for flow in pipes.

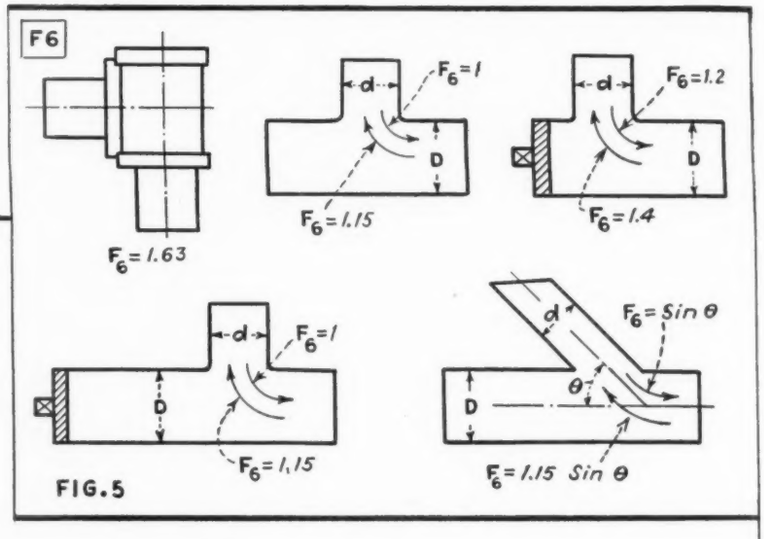


FIG. 5

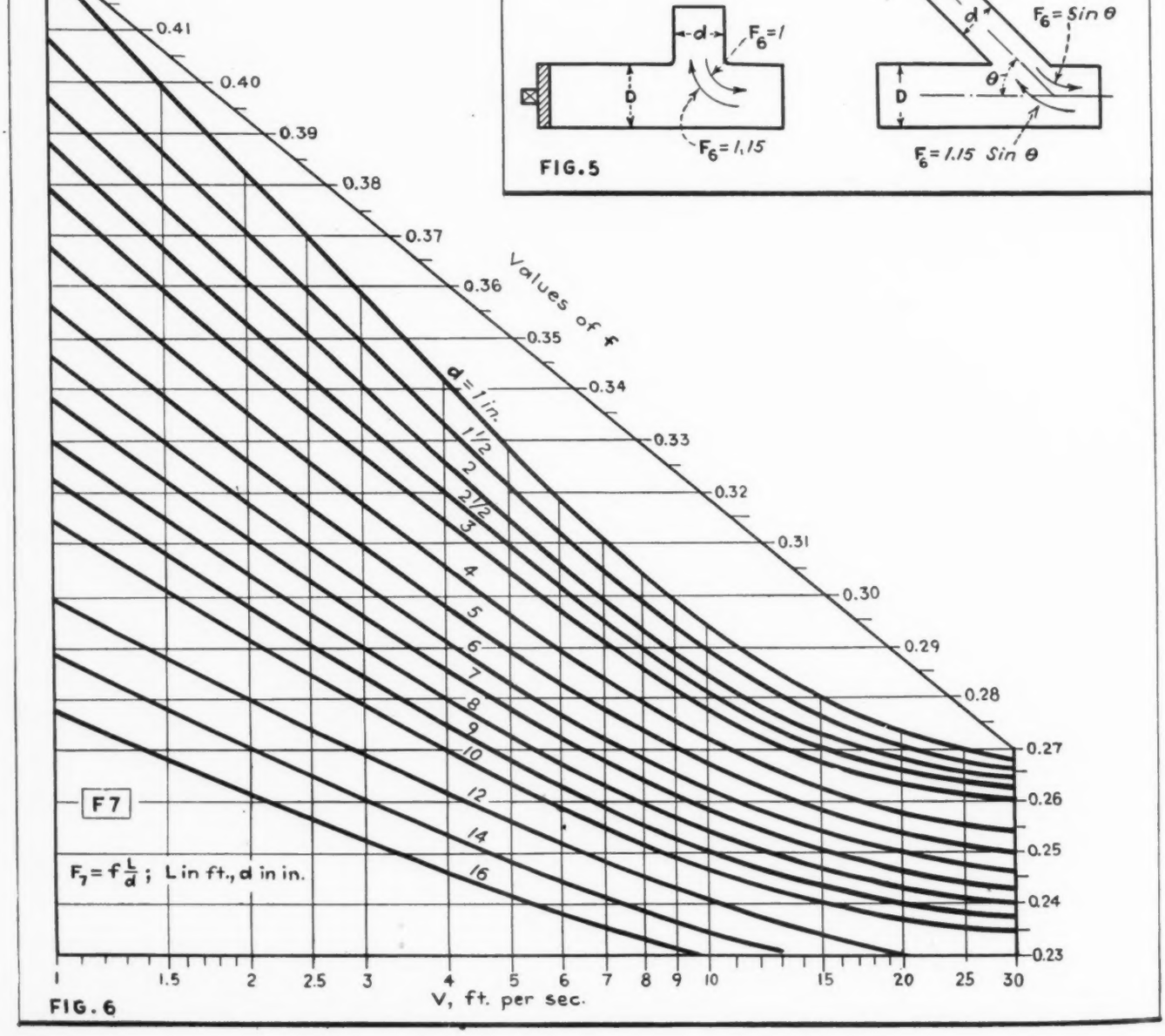


FIG. 6

8. Bends. In Fig. 7 values are given for a factor F_o for different velocities of flow and ratios of radius R of bend to pipe diameter d. Where

θ = angle of bend, deg.

the resistance coefficient F_8 of the bend is found from

$$F_8 = F_o\theta/90$$

9. Elbows. The resistance coefficients F_9 for non-reducing elbows are given in Table I for different angles of θ, where θ is the angle in degrees of the center line of the elbow.

10. Strainer Gratings of Perforated Sheets at Pipe Mouths.

A. Flat sheets. Resistance coefficients F_{10a} are given in Table II for different ratios of total passage area of holes s in flat perforated sheets to passage area of pipe S, both for flow from tank to pipe and for flow from pipe to tank.

B. Flat perforated sheet protecting the mouth of a gradually enlarged tube. Resistance coefficient is F_{10b}.

Where d = diameter of pipe
D = diameter of mouth

$$F_{10b} = F_{10a} \times \left(\frac{d}{D}\right)^4 + F_3$$

Table I—Resistance Coefficients F_9 for Elbows

θ, deg.	F_9	θ, deg.	F_9
30	0.11	70	0.58
40	0.20	80	0.90
50	0.32	90	1.13
60	0.47	120	1.90

Table II—Resistance Coefficients F_{10a} for Flat Perforated Sheet Strainers

s/S	Flow from tank to pipe F_{10a}	Flow from pipe to tank F_{10a}
0.45	3.95	6.48
0.50	3.04	5.16
0.55	2.41	4.22
0.60	1.97	3.50
0.65	1.63	2.96
0.70	1.37	2.52

Table III—Resistance Coefficients F_{10c} for Hemispherical Perforated Sheets

s/S	Flow from tank to pipe F_{10c}	Flow from pipe to tank F_{10c}
0.45	1.41	2.04
0.50	1.18	1.71
0.55	1.02	1.47
0.60	0.91	1.30

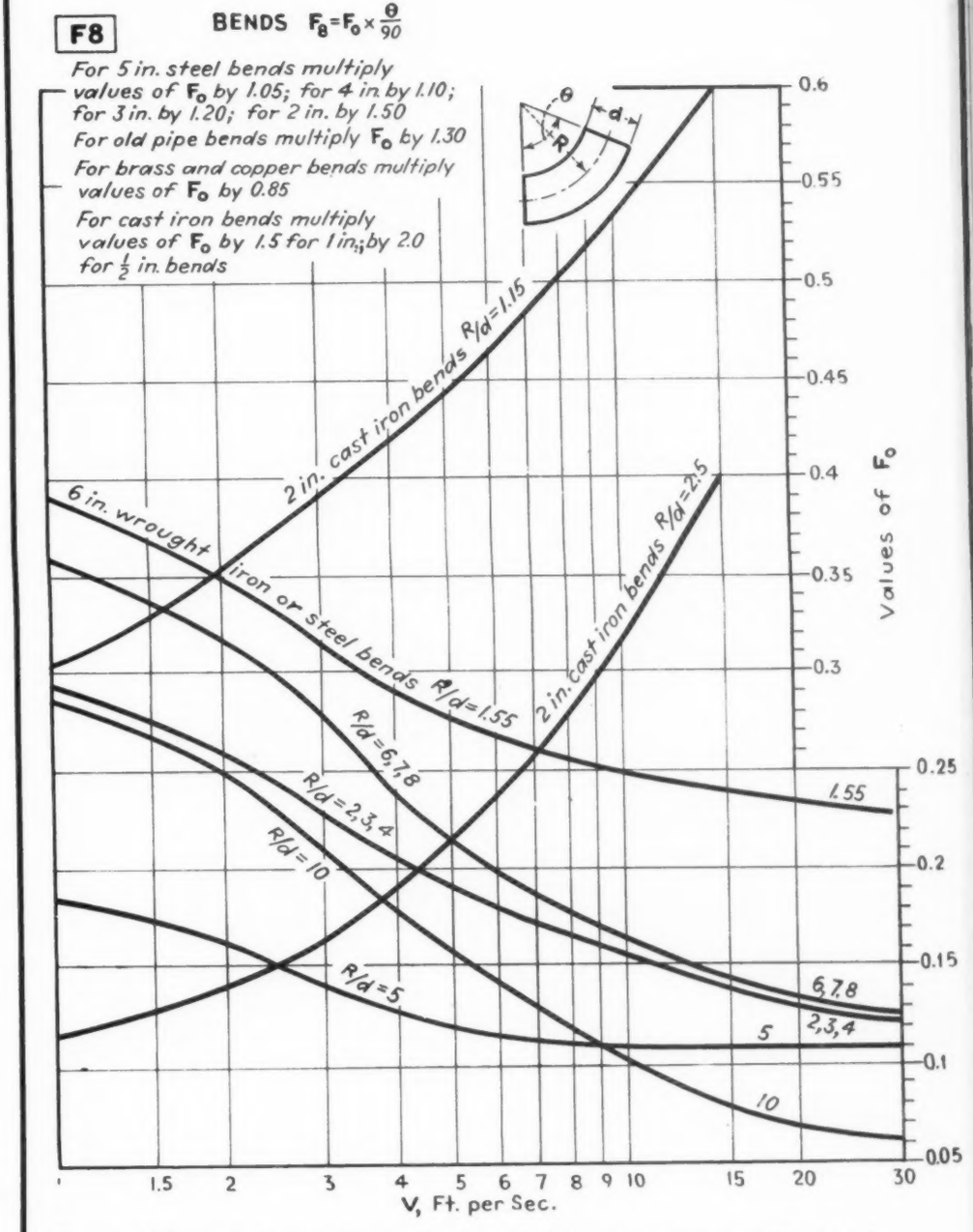


Fig. 7—Values of friction factor F_o for calculating resistance coefficients F_8 for pipe bends.

C. Hemispherical perforated sheets. The values for resistance coefficients F_{10c} for hemispherical perforated sheets are calculated with the aid of the data given in Table III. The values of F_{10c} listed in Table III have been estimated for a hemisphere having a diameter D equal to the diameter d of the adjacent pipe for different ratios s/S.

When the ratio D/d is greater than one, and the flow is from pipe to tank, the value of F_{10c} from Table III is divided by $(D/d)^4$, and to the value obtained is added the coefficient F_3 for sudden enlargement from Fig. 1. Thus

Corrected $F_{10c} = F_{10c}/(D/d)^4 + F_3$

When the ratio D/d is greater than one, and the flow is from tank to pipe, the value of F_{10c} from Table III is divided by $(D/d)^4$, and to the value obtained is added the coefficient F_4 for sudden contraction from Fig. 3, thus

Corrected $F_{10c} = F_{10c}/(D/d)^4 + F_4$

11. Gate, Globe and Angle Valves.

A, B, and C. In Table IV are given resistance coefficients F_{11a}, F_{11b}, F_{11c}, for gate, globe and angle valves respectively.

D. Cocks. $F_{11d} = 0.10$

E. Ferranti gate valves, fitted to 5 in. and larger pipes.

1. Passage area reduced to 25 percent, with D/d equal to 2, and convergent inlet and divergent outlet tubes of equal length, length being 0.7 D. F_{11e} equals 6.20.

2. Passage area reduced to 25 percent with length of inlet tube equal to 0.6 D, and length of outlet tube equal to 0.9 D. F_{11e} equals 3.80.

3. Passage area reduced to 43.3 percent,

with D/d equal to 1.52, length of inlet tube equal to 0.6 D, and length of outlet tube equal to 0.9 D. F_{11e} equals 1.05.

The following examples will illustrate how the data are applied.

Example 1. In the piping system shown in Fig. 8, the feed tank is 6 ft. wide by 10 ft. long, and

$H = 15$ ft.

Initial water level height in tank, $h_1 = 6$ ft.

Final water level height in tank, $h_2 = 1$ ft.

Ratio of pipe diameter d to jet diameter d_o is to be 2 or

$d/d_o = 2$

The strainer is a hemispherical perforated sheet the diameter D of which is 1.3 times the pipe diameter d, or

$D = 1.3d$

Total passage area of holes is equal to 50 percent of surface of hemisphere, or

$s/S = 0.50$

The bend has a center line radius R of 3.2 pipe diameter, or

$R/d = 3.2$

In the globe valve the pressure is on the disk.

The length L of the tapered portion of the jet is 4.2 times the jet diameter d_o, or

$L/d_o = 4.2$

Find the diameter of pipe and jet that will discharge the quantity of water between the levels specified in about 5 minutes.

Solution:

Mean theoretical velocity v_t of flow through the outlet orifice of jet is

$$v_t = \sqrt{2g\left(H + \frac{h_1 + h_2}{2}\right)}$$

$$= \sqrt{2 \times 32.2\left(15 + \frac{6+1}{2}\right)} = 34.4 \text{ ft. per sec.}$$

Volume of water to be discharged in 5 min. is

$$6 \times 10 \times (6 - 1) = 300 \text{ cu.ft.}$$

Discharge per sec. $q = \dfrac{300}{5 \times 60} = 1$ cu.ft.

Theoretical dia. d_t of jet in inches

$$d_t^2 = \frac{4 \times 1{,}728q}{12\pi v_o}$$

$$d_t = 13.53\sqrt{\frac{q}{v_o}} = 13.53\sqrt{\frac{1}{34.4}}$$

$$= 13.53 \times 0.171 = 2.314 \text{ in.}$$

Since losses will reduce the theoretical flow, the actual jet diameter will be larger than its theoretical diameter. For preliminary design purposes, select d_o equal to $2\frac{1}{2}$ in., and by means of the resistance coefficients given for the parts of the piping system determine what the actual flow would be.

With a jet diameter of $2\frac{1}{2}$ in., the pipe diameter d is 5 in.

For a discharge rate of 1 cu.ft. per sec. through a jet diameter of $2\frac{1}{2}$ in., the mean jet velocity v is

$$v = \frac{1 \times 4 \times 144}{\pi \times 2\frac{1}{2} \times 2\frac{1}{2}} = 29.4 \text{ ft. per sec.}$$

Assuming that the velocity of flow is the same in all parts of the 5 in. pipe line, the velocity in the parts is:

$$\left[\frac{\pi\,(2\frac{1}{2})^2}{4} \Big/ \frac{\pi\,5^2}{4}\right] \times 29.4 = 7.35 \text{ ft. per sec.}$$

Using a velocity of 7.35 ft. per sec. the local resistances are:

1. Strainer. From Table III and making necessary calculations for hemispherical strainer

$F_{10e} = 0.63$

2. Pipe. From Fig. 6, $f = 0.277$

then for a pipe 13.7 ft. long

$F_7 = 0.277 \times 13.7/5 = 0.76$

3. Bend. From Fig. 7

$F_o = 0.17 \times 1.05 = 0.179$

then $F_8 = 0.179 \times 90/90 = 0.179$

4. Globe Valve. From Table IV $F_{11b} = 4.90$

5. Jet. From Fig. 2, $\theta = 15$ deg.

From Fig. 3, $F_4 = 0.08$

6. Discharge Orifice. $F_{1e} = 1$

The sum of the resistance coefficients referred to the velocity of flow in the jet is

$$F_{1e} + F_4 + [F_{11b} + F_8 + F_7 + F_{10c}] \times (d_o/d)^4$$
$$= 1 + 0.08 + [4.90 + 0.179 + 0.76 + 0.63] \times \frac{1}{16}$$

$$= 1.08 + \frac{6.469}{16} = 1.08 + 0.404 = 1.484$$

The resistance coefficient for the system then is:

$$\mu = \frac{1}{\sqrt{1.484}} = \frac{1}{1.218} = 0.821$$

The mean velocity at jet is

$v_o = 0.821 \times 34.4 = 28.24$ ft. per sec.

Volume of discharge per sec. through $2\frac{1}{2}$ in. jet is:

$$q = \frac{\pi\,(2\frac{1}{2})^2 \times 28.24}{4 \times 144} = 0.965 \text{ cu.ft. per sec.}$$

Time to discharge tank

$$\frac{300}{0.965 \times 60} = 5.2 \text{ min.}$$

If this time does not satisfy problem specifications, repeat calculations using next larger size pipe.

Example 2. Pipe line to be the same as for Example 1, except the jet is removed and replaced with a short nipple. All other specifications to be the same as for Example 1. Find diameter of pipe.

Solution:

Mean theoretical velocity v_t of flow through nipple is 34.4 ft. per sec.

For preliminary design purposes, select d_o equal to

$(2\frac{1}{2} + 5)/2 = 3.75$ in. or roughly 4 in.

then establish resistance coefficients.

For a discharge rate of 1 cu.ft. per sec. through a 4 in. pipe and nipple, the mean actual velocity should be

$$V = \frac{1 \times 4 \times 144}{\pi \times 4 \times 4} = 11.45 \text{ ft. per sec.}$$

Using a velocity of 11.45 ft. per sec. the local resistances are:

1. Strainer. From Table III and making necessary calculations for hemispherical strainer

$$F_{10e} = \frac{1.18}{1.3^4} + F_3$$

From Fig. 1, $F_3 = 0.22$

$$F_{10e} = \frac{1.18}{2.86} + 0.22 = 0.413 + 0.22 = 0.633$$

2. Pipe. From Fig. 6

$f = 0.269$

Table IV—Resistance Coefficients F_{11a}, F_{11b} and F_{11c} for Valves

Working faces 45 deg. to full opening

Size, in.	1	1½	2	2½	3	4	5	6	7	8	10	12
F_{11a}, Gate valves	0.13	0.12	0.11	0.11	0.10	0.10	0.09	0.08	0.08	0.07	0.07	0.07
F_{11b}, Globe valves												
Pressure on disk	9.18	8.30	7.30	6.80	6.27	5.50	4.90	4.55	4.20	3.95	3.58	3.25
Pressure under disk	9.77	9.05	8.12	7.83	7.30	6.55	5.86	5.68	5.38	5.20	4.85	4.50
F_{11c}, Angle valves												
Pressure on disk	3.65	3.12	2.75	2.52	2.30	2.00	1.81	1.76	1.52	1.45	1.35	1.30
Pressure under disk	4.00	3.50	3.16	2.97	2.77	2.45	2.30	2.28	2.00	1.94	1.83	1.78

Note: In general, the coefficient for a globe valve is 50 times that for a full-flow gate valve or a cock.

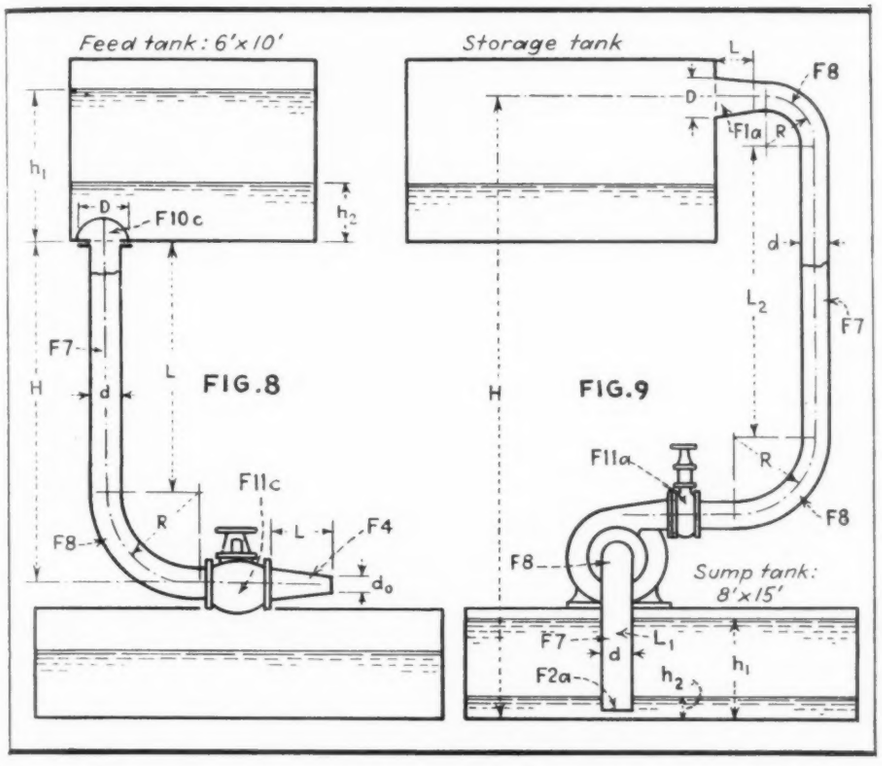


Fig. 8—Pipe system for emptying a feed tank by gravity flow. Fig. 9—Pipe system with centrifugal pump for filling an overhead storage tank.

then for a pipe and nipple 14 ft. long

$F_7 = 0.269 \times 14/4 = 0.943$

3. Bend. From Fig. 7

$F_6 = 0.15 \times 1.1 = 0.165$

then $F_8 = 0.165 \times \frac{90}{90} = 0.165$

4. Globe Valve. From Table IV, $F_{11b} = 5.50$

5. Discharge at nipple, $F_{1a} = 1$

$$\Sigma F = F_{1a} + F_{11b} + F_8 + F_7 + F_{10c} = 1 + 5.50 + 0.165 + 0.943 + 0.633 = 8.241$$

The resistance coefficient for the system then is

$$\mu = \frac{1}{\sqrt{8.241}} = 0.35$$

The mean velocity in 4 in. pipe is

$$v = \frac{34.4}{\sqrt{8.241}} = 11.97 \text{ ft. per sec.}$$

Volume of discharge per sec. through 4 in. pipe

$$q = \frac{\pi (4)^2 \times 11.97}{4 \times 144} = 1.04 \text{ cu.ft. per sec.}$$

Time to discharge tank

$$\frac{300}{1.04 \times 60} = 4.8 \text{ min.}$$

Example 3. In Fig. 9 is shown a centrifugal pump delivering water to a storage tank. All pipe and fittings are 3 in. size.

Height of discharge orifice in side of overhead storage tank from bottom of sump tank

$H = 40$ ft.

Sump tank is 15 ft. long by 8 ft. wide.

Initial water level in sump tank, $h_1 = 8$ ft.

Final water level in sump tank, $h_2 = 1$ ft.

The capacity of the pump is to be such that water in the sump tank between the levels specified can be pumped out in about 10 min. Pump efficiency is 60 percent.

Radius of center line of pipe bends is to be 3 times their diameter.

Mouth of discharge pipe is 1.3 times pipe diameter, length of taper at enlargement is 1.4 pipe diameter.

Length of discharge pipe between bends, $L_2 = 20$ ft.

Straight length of suction pipe, $L_1 = 10$ ft.

Valve is the gate type.

Find horsepower required to drive pump.

Solution: Theoretical velocity of flow corresponding to mean pressure head:

$$v_t = \sqrt{2g\left(H - \frac{h_1 + h_2}{2}\right)} = \sqrt{2 \times 32.2\left(40 - \frac{8+1}{2}\right)} = \sqrt{2{,}280} = 47.8 \text{ ft. per sec.}$$

Volume of water to be pumped out of sump tank

$15 \times 8 \times (h_1 - h_2) = 120 \times 7 = 840$ cu. ft.

Discharge per second

$$q = \frac{840}{10 \times 60} = 1.4 \text{ cu.ft. per sec.}$$

Mean velocity through a 3 in. pipe

$$v = \frac{1.4 \times 4 \times 144}{\pi\, 3^2} = 28.45 \text{ ft. per sec.}$$

The local resistance coefficients are:

1. Suction Pipe Inlet. $F_{2a} = 0.49$
2. Three Pipe Bends. From Fig. 7 $F_6 = 0.123$ $F_8 = 0.123 \times 90/90 \times 3F_8 = 0.4428$ $1.20 = 0.1476$
3. Gate Valve. Table IV, $F_{11a} = 0.10$
4. Pipe. From Fig. 6, $f = 0.26$

then for a pipe 10 + 20 or 30 ft. long

$F_7 = 0.26 \times 30/3 = 2.60$

5. Enlargement of Discharge Pipe. From Fig. 2, $\theta = 12.5$ deg. From Fig. 1, $F_3 = 0.04$
6. Discharge Orifice.

$F_{1a} = 1 \times (1/1.3)^4 = 0.35$

$\Sigma F = 0.49 + 0.443 + 0.10 + 2.60 + 0.04 + 0.35 = 4.023$

Where

w = weight of water per cu. ft. = 62.4 lb.
P = pump delivery pressure, lb. per sq. in.
g = 32.16 ft. per sec. per sec.
V_d = velocity of discharge, ft. per sec.
V_t = mean theoretical velocity of flow, ft. per sec.

the fundamental hydrodynamic equation is

$$-\frac{V_t^2}{2} = -2g\frac{144\,P}{w} + \frac{v^2}{2} \quad \Sigma F$$

Then by substitution

$$P = \frac{47.8^2 + 28.45^2 \times 4.023}{296} = 18.7 \text{ lb. per sq. in.}$$

Horsepower required to drive pump

$$\frac{18.7 \times \pi \times 3^2 \times 28.45}{4 \times 550 \times 0.6} = 11.4 \text{ hp.}$$

SOURCES OF DATA used as a basis for establishing the coefficients of resistance here cited or given in chart form are:

For F_1 and F_2, Smith's experiments, see "Treatise on Hydraulics", Saph and Schoder.

For F_3, *Proceedings* of the Royal Society.

For F_4 and F_5, Freeman's and Flienger's experiments

For F_6, *Transactions American Society of Civil Engineers*, Vol. 62, together, with Prof. Gibson's experiments.

For F_7, Fanning's experiments, see "Treatise on Hydraulics", Saph and Schoder.

For F_8, *Transactions American Society of Civil Engineers*, Vol. 62. Davies' experiments in Wisconsin.

For F_9, and F_{10}. These are calculated on the basis of F_3, F_4, F_5, F_6, and controlled by the author's experiments.

For F_{11}, established by experiments made by Vicker's Navy Yard and the Hydraulic Laboratory of St. Petersburgh Politechnicum.

INDUSTRY AND SOCIETIES

RESEARCH—STANDARDS—PATENTS

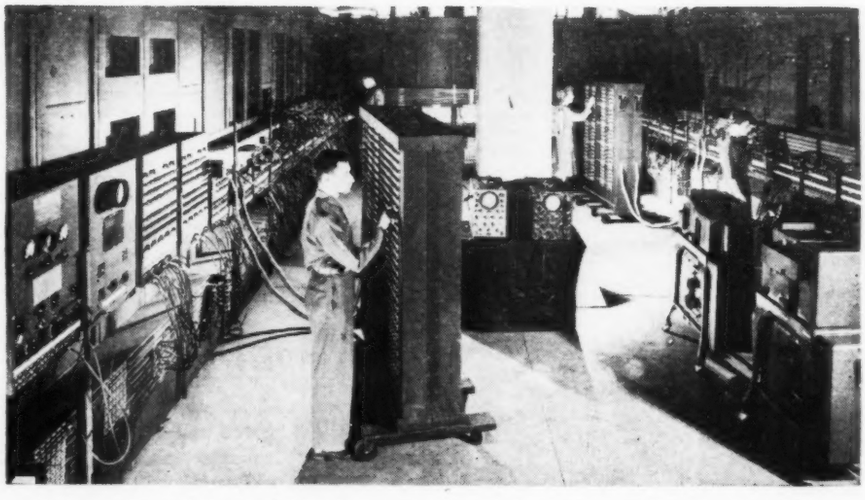
Electronic Numerical Integrator and Computer

New Machine to Aid Industrial Design

A NEW MACHINE that is expected to revolutionize the mathematics of engineering and change many of our design methods has been developed by the Army Ordnance Department.

Designed and constructed for the Ordnance Department at the Moore School of Electrical Engineering of the University of Pennsylvania, this machine is the first all-electronic general purpose computer ever developed. It is capable of solving many technical and scientific problems so complex and difficult that all previous methods of solution were considered impractical.

The mathematical robot, known as the ENIAC (Electronic Numerical Integrator and Computer), is the invention of Dr. J. W. Mauchly and Mr. J. Presper Eckert, Jr., both of the Moore School. Begun in 1943 at the request of the Ordnance Department to break a mathematical bottleneck in ballistic research, its peacetime uses extend to all branches of science and engineering.



Overall view of the ENIAC, being prepared for a hydrodynamical problem.

Fortunately for our modern mode of living many design problems were possible of solution by ordinary mathematical means and in a "practical" length of time. The introduction of the differential analyzer expanded the scope of physical problems which could be readily solved by mechanical means. This type of computing device which is sometimes referred to as a continuous variable machine will accept a fairly large class of total differential equations. The a.c. network analyzer further expanded the scope of problems which could be solved. However, the class of problems which can be solved by both these means is still very limited. Further, since these machines perform all operations of a computation in parallel, the size and complexity of the specific problem they can solve, even in the class which they will accept, is limited by their number of arithmetic components.

Need for High Speed Computers

Physical problems that cannot be solved analytically (i.e., cannot be solved by formulas) are increasing in number. Such problems have been handled by computational methods or through the use of specific analogy machines. An illustration of the computational approach is the work of the English physicist, Douglas R. Hartree, on the structure of the atom. This work involved a series of calculations extending over a period of 15 years, by the methods available to Hartree.

The new complexity of problems arising in the physical sciences have made specialized devices necessary. Just as higher mathematical systems had to be invented to deal with theories which could no longer be handled by ordinary arithmetic, so new devices to make these theories of practical value had to be designed. An interesting, yet comparatively simple, example of such specialized devices is the planimeter which works by means of a mechanical analogy; the relationship between a cam and disk represents the formula, which is used to find the area under a curve.

A further example of the analogy technique is found in the use of wind tunnels. At present, the supersonic wind tunnel at the Ballistic Research Laboratory at Aberdeen is used about 30 percent of the time as an analogy machine to solve aerodynamical problems. Industrial research organizations frequently make use of highly specific analogy machines, such as the a.c. network analyzer mentioned previously, to solve such problems as electric circuit theory equations or the partial differential equations that enter into electron-optics problems. For the latter, thousands of rubber analogy "models" are built every year. Not only are such techniques lengthy and costly, but a sufficient degree of accuracy is not always obtained.

Continuous and Discrete Variables

There is a fundamental difference between the planimeter, for example, which is a continuous variable type of calculating device and arithmetic devices which operate on discrete variables or in a step-wise fashion.

The first suggestion for a large scale continuous variable machine, now called a differential analyzer, was made by the English physicist, Lord Kelvin, in a series of papers given before the Royal Society in 1876. Machines of this kind had been built at least five different times before one of practical design was produced in this country.

The first successful differential analyzer was designed and built in this country in 1925. The designer was Dr. Vannevar Bush, distinguished wartime head of the Office of

Scientific Research and Development, who was at that time Professor of Electrical Engineering at the Massachusetts Institute of Technology where the machine was built. A more comprehensive differential analyzer was built by Dr. Bush and his associates at M.I.T. in 1930. In the Fall of 1945 another differential analyzer of the Bush type was announced at M.I.T. This machine includes no basic principles which were not present in the earlier machine, except that various elements are coupled electrically and the data fed in by special tapes.

Although the M.I.T. analyzer and others of its kind, including the two used by the Ballistic Research Laboratory of the U. S. Army Ordnance Department were doing work throughout the war, it became increasingly evident that differential analyzers and other existing continuous and discrete variable machines had shortcomings which made it impossible to keep pace mathematically with new physical discoveries. In making up ordnance firing and bombing tables for instance, the differential analyzer can perform many labor-saving operations up to a certain point, but beyond that, individual calculations must be performed.

Besides not being able to handle discrete calculations, differential analyzers cannot work practicably with high-order or partial-differential equations. A typical problem of this type involves shock-wave calculations in the design of ordnance projectiles.

The obvious limitations of existing devices of both the continuous variable and discrete variable types indicated that high-speed digital calculators, capable of handling all types of computing problems, were urgently needed. The first such large scale general purpose machine was built by the International Business Machines Corporation and dedicated at Harvard University in 1944. This machine, called the Automatic Sequence Controlled Calculator, can perform many operations not possible on a differential analyzer. The problems of speed and flexibility were by no means completely solved by the Harvard machine. Although elaborate sequences of operations are completed fairly rapidly, limitations imposed by the action times of mechanical parts make its overall speed much less than that of a purely electronic device. It remained for the ENIAC to make complicated design calculations, such as the projectile design problem mentioned previously, possible in a practical length of time.

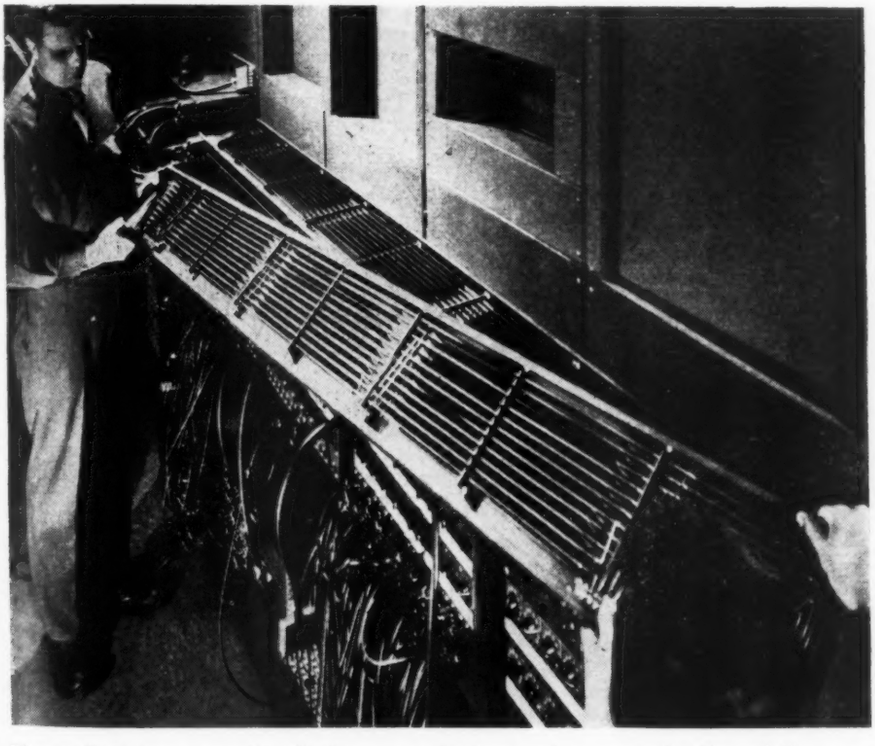
Description

The ENIAC occupies a room 30 by 50 feet in size. It weighs 30 tons and has 100 feet of front panels. This machine is the most intricate and complex electronic device in the world, requiring for its operation 18,000 electronic tubes There are no moving parts in the machine.

External to the ENIAC, but a part of the total installation are two comparatively small I.B.M. punch-card machines which feed information into the ENIAC from punched cards and receive the results therefrom in a similar manner.

ENIAC Timing Chart

Operation	Micro-sec.	Addition Times
Addition, Subtraction, or Transfer		
Including transfer from constant transmitter	200	1
Transfer repeated n times in the accumulator (n = 1 to 9)	$200n$	n
Multiplication		
By n digit multiplier (n = 2 to 10)	$800 + 200n$	$4 + n$
By ten digit multiplier	2,800	14
Division or Square Rooting		
Average time for nine digit result	26,000	130
Average time for n digit result (n = 3, 6, 7, 8, or 9)	$(n + 1) \times 2600$	$13 \times (n + 1)$
Maximum time for nine digit result	42,000	210
Maximum time for n digit result (n = 3, 6, 7, 8, or 9)	$(n + 1) \times 4200$	$21 \times (n + 1)$
Obtaining a Functional Value Once	1000	5
Repeated n times (n = 1 to 9)	$800 + 200n$	$4 + n$



These digit trays consist of eleven conductors and a ground. Ten of the conductors represent the ten digits, maximum number of significant figures handled by the machine. The eleventh conductor carries the sign indication.

The forty main panels of the ENIAC are arranged in a large U, with 16 of the panels on each leg and 8 panels on the end. (See layout diagram on the following page.)

The ENIAC is the fastest computing machine in the world, and is 500 times faster than any existing machine. It is about 1000 times as fast as the Harvard-I.B.M. computer. Time consumed in basic operations is listed in the accompanying "ENIAC Timing Chart."

The ENIAC consumes 150 kilowatts. This power is supplied by a three phase regulated, 240 volt, 60 cycle power line. The power consumption may be broken up as follows: 80 kilowatts for heating the tubes, 45 kilowatts for generating d.c. voltages, 20 kilowatts for driving the ventilator blower and 5 kilowatts for the auxiliary card machines.

The operational characteristics of the ENIAC may be classified as follows: arithmetic, memory and control elements.

The arithmetic elements of the ENIAC exist in a number of units listed in the

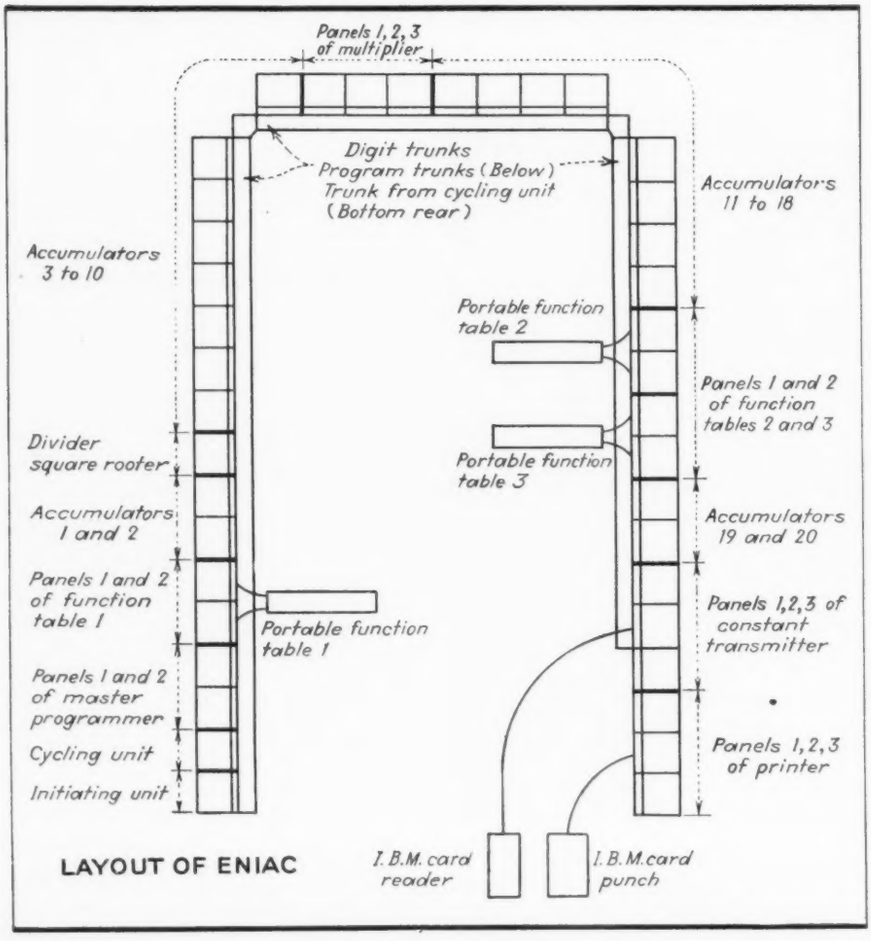
accompanying layout diagram. The most important of these are: 20 accumulators, 1 multiplier and 1 combination divider and square rooter. The accumulators provide means for storing numbers computed in the course of a problem and for allowing the addition or subtraction of a second number to or from the stored number. They are capable of performing these operations on numbers having as many as ten decimal digits and indicate the associated plus or minus sign. The multiplier can find the product of two demical numbers, each having as many as ten digits. The combination divider and square rooter can find the quotient of two nine-digit decimal numbers or the square root of a nine-digit decimal.

Memory elements of the machine can be divided into two groups: "internal memory" and "external memory." The former exist in all memory devices within the machine, thus being of limited capacity, while the latter exist outside of the machine in the form of punched cards and is accordingly of unlimited capacity. Communication between the internal and external memory takes place at speeds which are set by the mechanical punch card devices.

How It Works

Control elements include the initiating unit, concerned mainly with starting and stopping, and the cycling unit which generates the fundamental signals used in the ENIAC. Since the ENIAC contains a number of trunk circuits and because all units are synchronized by permanent electrical connections with the cycling unit, operations between pairs of ENIAC units can be carried out simultaneously. The cycling unit contains an oscillator which generates electrical impulses at the rate of 100,000 per second, each pulse having a duration of two microseconds (millionths of a second). These pulses are fed into a 20-position electronic stepping switch or "counter," which enables the cycling unit to put out a special impulse or "program pulse" at every 20th pulse of the "clock" (oscillator) and a train of other fundamental pulses during the interval between program pulses. The program pulses form the basis of the programming system and set off the beginning and end of the addition cycles which are the basic arithmetical intervals of the machine. The addition cycles thus are repeated at 1/20 of the "clock" rate or at the rate of 5000 per second. The addition cycle or addition time, 1/5000 of a second or 200 micro seconds, is so named because one such cycle is required to complete an addition in the ENIAC.

The general method of handling a problem with this machine is as follows: First, the engineer must analyze his problem so that he can write down mathematical equations that express the phenomena involved. Second, he must break down the mathematical formulation into a sequence of additions, subtractions, multiplications, divisions, square rootings and transfers from unit to unit of the ENIAC so that the ENIAC can be instructed to make the computations. Finally, a small amount of time is required in preparing the ENIAC for a problem by such steps as setting program switches, putting numbers into the function table memory by setting its switches, and establishing connections between units of the ENIAC for the communication of programming and numerical information.


Panels 1, 2, 3 of multiplier
Digit trunks
Program trunks (Below)
Trunk from cycling unit (Bottom rear)
Accumulators 11 to 18
Accumulators 3 to 10
Portable function table 2
Panels 1 and 2 of function tables 2 and 3
Portable function table 3
Divider square rooter
Accumulators 1 and 2
Accumulators 19 and 20
Panels 1 and 2 of function table 1
Portable function table 1
Panels 1,2,3 of constant transmitter
Panels 1 and 2 of master programmer
Cycling unit
Initiating unit
Panels 1,2,3 of printer
I.B.M. card reader
I.B.M. card punch
LAYOUT OF ENIAC

Industrial Applications

The electrical and electronics industries are among the first who may make use of the ENIAC-type calculators. Design of much complex and expensive equipment such as transformers, rectifiers, generators, motors and other heavy-duty units is limited by many unknown factors. Model-building on a large scale is too expensive; analogy machines, such as network analyzers, are used. The use of such machines cannot be considered true experimentation; their parts "represent" certain physical quantities, and often no close relationship exists between variation in these quantities and change in materials or shape of product parts. Reasoning by analogy is an accepted logical instrument, but drawing analogies should not replace scientific "cause-and-effect" thinking. Moreover, these analogy devices represent a large dollar-and-cents investment, especially when one considers the narrowness of scope of the problems they can accept.

In the electronic industry, rubber models are used as analogy devices in the designing of vacuum tubes in order to determine the electro-magnetic characteristics involved. These are not really models in the true sense of the word: tiny steel balls "represent" electrons; curved paths "represent" various quantities, such as potential energy.

Design of mechanical products, especially prime movers, is likewise a costly process. Electrical analogy machines are also used in this field: mechanical problems are translated into electrical analogies, and then back again. Expensive mechanisms such as steam and gas turbines, diesel engines and jet engines do not lend themselves to model-building because of costs. Here again, a high-speed general purpose computer is a definite need.

Aircraft design and research are extremely costly processes, because of the need for wind tunnel experimentation. The designers of the ENIAC believe that from 30 to 50 percent of this type of experimentation could be eliminated by the use of high speed calculation. The entire field of hydrodynamics, or the mechanics of fluids in motion, is becoming prohibitively costly, and is sorely in need of mathematical treatment.

No one expects that the industrial effects of the ENIAC will become far-reaching overnight; it is felt, however, that large industries which mass-produce heavy and expensive products, will be among the first to make use of the ENIAC, at least for the research phases of design.

Stationary Gas Turbine Operates at 1500 Degrees F.

A 3,500-HP. EXPERIMENTAL GAS TURBINE capable of operating at temperatures up to 1,500 deg. F. has been designed and built for the Navy by the Allis-Chalmers Mfg. Company. Although it has been tested for two years at the U. S. Naval Engineering Experimental station at Annapolis, publicity concerning this unit was not permitted until recently.

A modern high temperature steam plant operates at exhaust temperature, and an inlet temperature of 950 deg. F. Gas turbines must operate at a terminal temperature from 500 to 1,000 deg. F. at atmospheric pressure, since the exhaust gas from an open cycle gas turbine cannot be condensed. For comparable and better efficiency than steam plants, it is clear that inlet temperatures must be considerably higher than 950 deg. F.

Although gas turbine-driven superchargers for bombers and fighters during the war operated up to temperatures of 1,600 deg. F., the present outlook for the application of gas turbines for marine propulsion and other power sources is operation at 1,500 deg. F. The aim of the Annapolis unit is to guide designers of future multiple-stage gas turbines operating at this temperature.

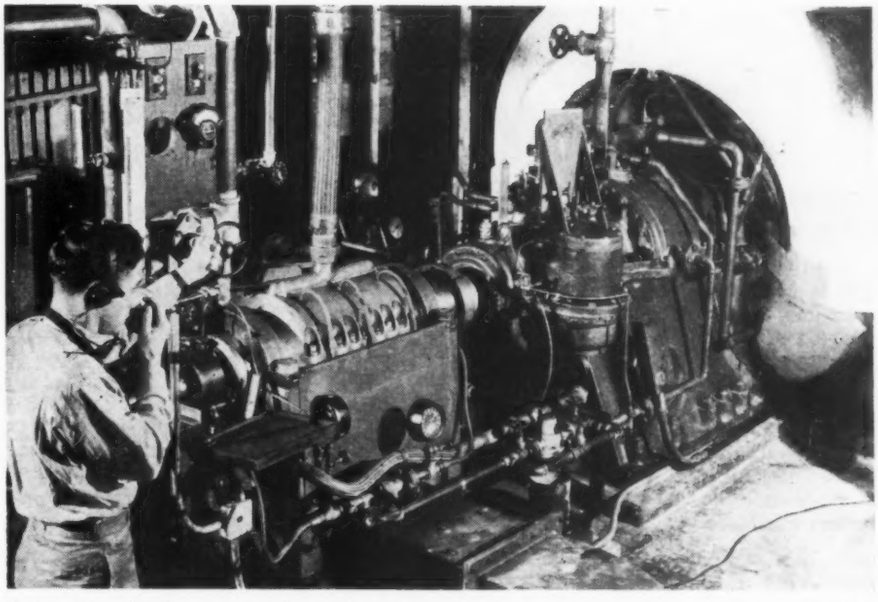
Metallurgical composition of the parts of the Allis-Chalmers-Navy turbine, as in the case of war-born turbosuperchargers and jet propulsion units, is not disclosed. It is clear that, for reliable design of these parts, composition is the all-important factor.

One design modification, however, made necessary by the nature of the new alloys, is revealed. This is cooling of the high temperature disks at the high pressure ends of the turbines. The Annapolis turbine includes such cooling features for the high temperature spindle parts.

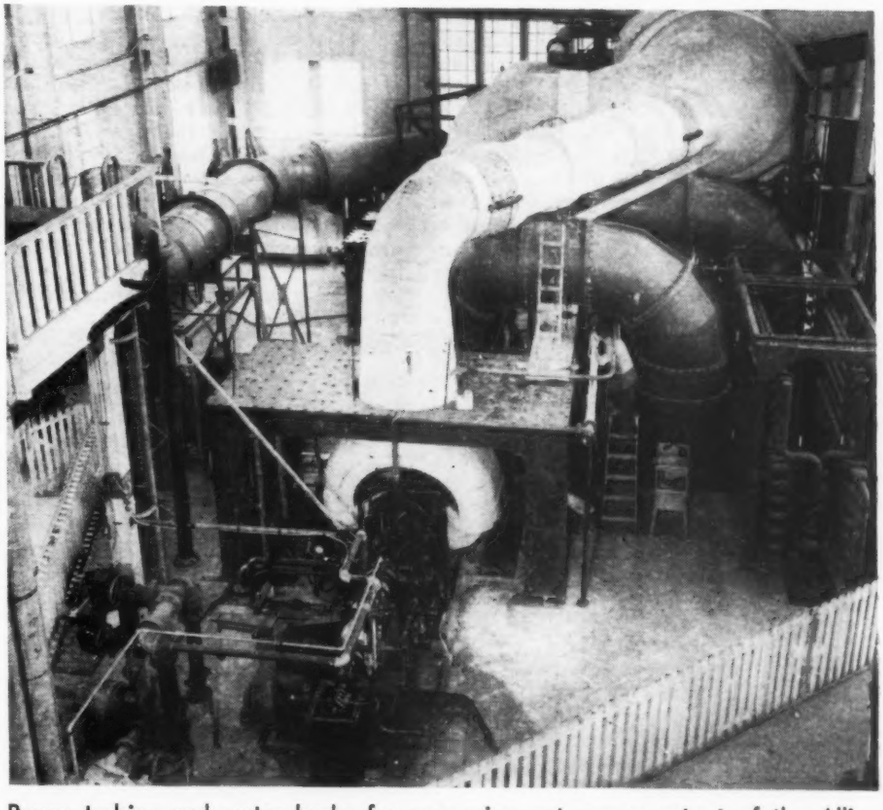
Special provision for the linear expansion of piping at these high temperatures were required. A special type of expansion joint is used at several points in the installation; this accommodates the expansion by the rotation of the adjoining members of the joint. This piping was designed with an inner skin or gas conducting member, over whose walls there is no pressure difference. Mineral block insulation between the inner and outer wall establishes the temperature drop, whereby the outer wall is cooled sufficiently to maintain physical strength.

The Cycle

The cycle selected for the Annapolis turbine is known as the parallel turbine regenerative cycle. One turbine drives the gas generator compressor and the second, or power turbine drives the water brake or dynamometer, which absorbs the output while measuring it at the same time. Air enters the gas generator compressor at atmospheric pressure, and is there compressed to 45 lb. per sq. in. gage pressure. The compressed air then enters the heat exchanger, where it is heated by the turbine exhaust gas from a temperature of 363 deg. F. at the



Exhaust end of power turbine (right) is point where net power output of entire unit can be measured as it is absorbed by water brake (center).



Power turbine and water brake for measuring net power output of the Allis-Chalmers experimental gas turbine at Annapolis is shown above in the foreground. At upper right is the regenerator, or heat exchanger, which preheats air passing from compressor to the two combustion chambers. This heat is salvaged from the hot exhaust gases leaving the two gas turbines. At right center is a battery of coolers for the air used in internal turbine cooling.

compressor discharge to a temperature of 750 deg. F. at the entrance to the combustion chambers. After leaving the heat exchanger, the preheated air flows directly downward into the two horizontal combustion chambers, one of which supplies the gas generator turbine driving the compressor, the other supplying the power turbine. Power generated by the gas generator and power turbines can be independently controlled.

Simplified Practice Recomendations

PRINTED COPIES of Simplified Practice Recommendation R212-45, Cast-Brass Solder-Joint Fittings, are now available, according to an announcement of the Division of Simplified Practice, National Bureau of Standards. The recommendation sets up a stock list of east-brass solder-joint fittings representing the best thought of the industry, its distributors and customers as to what constitutes desirable practice for the immediate future. Sketches illustrate the 37 types of fittings, various sizes of which are included in the list. Copies of Simplified Practice Recommendations R212-45 may be obtained from the Superintendent of Documents, Government Printing Office, Washington 25, D. C., for 5 cents each.

* * *

A proposed Simplified Practice Recommendation covering automatic regulating valves, which was submitted to producers, distributors and users last November by the National Bureau of Standards, has been approved.

Automatic regulating valves are used in industry for controlling temperature, pressure, level or rate of flow of fluids. The purpose of the recommendation—R219-46, Automatic Regulating Valves—is to establish as a useful standard of practice in the industry a simplified list of pressure ratings and sizes of seven types of valves made of steel, iron and bronze.

It was developed by a simplified practice committee of manufacturers in cooperation with the Division of Simplified Practice of the Bureau of Standards. The recommendation will retain on a voluntary basis certain of the simplified practice features of a mandatory issued by the War Production Board which was effective during the war.

* * *

A proposed Simplified Practice Recommendation for Steel Rivets has been submitted to producers, distributors and users for consideration, comment and approval. Mimeographed copies of the proposed recommendation may be obtained from the Division of Simplified Practice, National Bureau of Standards, Washington 25, D. C.

* * *

A voluntary Simplified Practice Recommendation for Hot-Rolled Carbon-Steel Bars and Bar-Size Shapes, (Billet Steel Mill Practice), as proposed by the Technical Committee on Carbon Steel Bars, of the American Iron and Steel Institute, has been submitted to all interests, for their comment, acceptance as proposed, or both.

The proposal covers the nominal sizes of: rounds; squares; round-corner squares; hexagons; half-rounds; ovals; half-ovals; bar-sizes of angles, channels and tees. Readers of this announcement are invited to make known their interests.

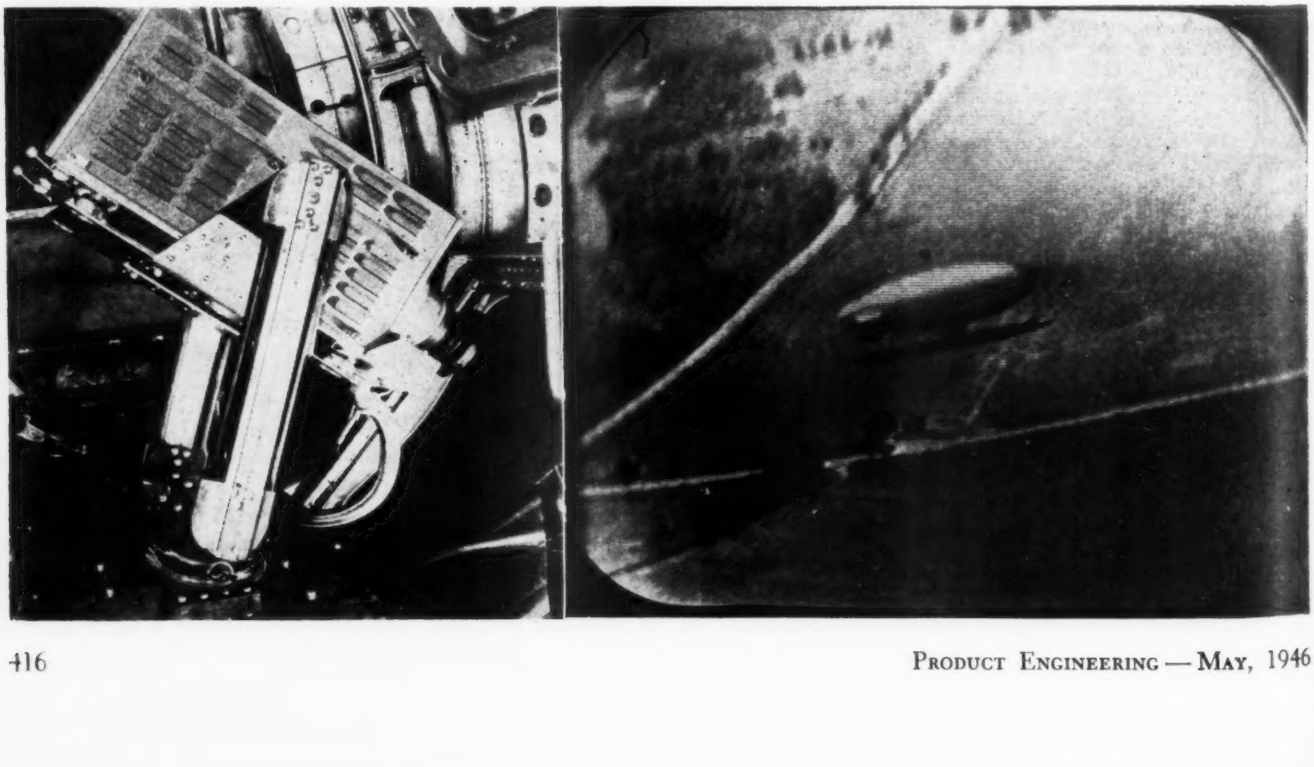
Airborne Television Has Commercial Future

A SUITCASE SIZED aerial television camera was among latest television equipment demonstrated by the Navy at the Anacostia Naval Air Station Developed for combat reconnaissance and guided-missiles, the equipment was divided into two classifications. "Block" and "Ring" television. Designed by the Navy and the Radio Corporation of America, it was built by R.C.A. and the National Broadcasting Company.

Television pick-up and transmitting equipment that once might have filled a large room was redesigned and built to "suitcase" compactness for military use in the Block system. The camera weighs about 34 lb., and the transmitter, monitor, power supply and junction box add 90 lb. to the installed weight. "Block" has a range of 15 to 20 miles, a peak power output of 60 watts at 264 to 372 megacycles, and produces 40 frames a second in sequential scanning, with 350 lines on the receiving screen. Special transmitting antennas have been designed for each of its ten workable channels. It may be a forerunner of portable "walkie-lookies" for covering news events, and may be used for industrial remote control.

A large version, called "Ring," was developed too late for use in the war. Installed, it weighs 1,400 lbs. with all accessories. "Ring" has a transmitting radius of 200 miles at an altitude of 22,500 ft. Its peak power output is 1.4 kw. of 90 or 102 megacycles, and a specially designed antenna gives uniform radiation in all directions. It uses two movable cameras, while "Block" is fixed in the nose of the plane.

Photograph at left shows a "Ring" camera installed at the waist hatches of a JM-1 Marauder. At right a moored blimp as seen on a "Ring" receiving scope eight miles from the plane, at an altitude of 1,000 ft.



WASHINGTON NOTES

Government to Enter Into Engineering Research Field

Gradually, the administration's policy on technical research is taking shape, emerging as the clear-cut, unadulterated program of the Kilgore-Henry Wallace camp in embattled Washington. Readers of this column are by now familiar with the Kilgore "compromise" bill for support of basic scientific research. Containing ideas specifically opposed by all the major engineering societies, this piece of administration-sponsored legislation is soon to be introduced to an unsuspecting Senate.

A comparatively unknown side of the administration's technical "aid" program however, kept warming up on the sidelines until the Kilgore-Magnuson squabble was safely settled, is about to enter the race for Congressional passage. This is the bill sponsored by Senator J. W. Fulbright of Arkansas, S. 1248, which would establish an "Office of Technical Services" in the Commerce Department.

U. S. Army Air Forces

NOW IN PROCESS OF ASSEMBLY is the Army's newest super bomber, the XB-36, a six-engined pusher type plane, described by General H. H. Arnold as larger, faster and more powerful than the B-29 Superfortress. It is shown above in artist's sketch.

Covering the Entire Field—In the report issued by the prolific and numerous Kilgore staff in connection with his "compromise" bill, S.1850, the administration's plan for "total" federal coverage of science and engineering is revealed. Talking about the functions of the various divisions of the National Science Foundation which would be set up under the Kilgore bill, the report says: "The subcommittee (Kilgore's Subcommittee on War Mobilization) conceives the function of the Division of Engineering and Technology to be the support of research in the fundamental engineering sciences and other studies basic to the broad development of technology, not the engineering development of machines or processes. It is thus clear that the functions of this division do not conflict with those of the Office of Technical Services which S.1248 would establish in the Department of Commerce to aid private inventors and industries with the engineering development of new ideas."

The phrase "engineering development of machines or processes" lets the cat out of the bag where the Fulbright bill is concerned. Is it possible, Washington is asking, that Kilgore and Fulbright have divided the entire field of basic and applied science between them, tying up government control into two neat little legislative packages?

The Fulbright Bill—Shortly to be introduced on the floor of the Senate, S. 1248 has two avowed purposes. Its primary aim is to "coordinate the Government's present technological research facilities and information to make them more readily available to business." In plain language, this means that various government agencies, permanent and war-born, would be lumped together in a super technical "service" office in the Commerce Department. Agencies in the new office would include the National Inventors' Council, Office of the Publication Board, the Office of Production Research and Development, formerly part of the War Production Board and the Technical Advisory Service of Maury Maverick's now-defunct Smaller War Plants Corporation. The inclusion of this last agency has drawn most fire from engineering circles, where the feeling is that Mr. Maverick's agency made a complete fiasco of technical service and no reason can be discerned for such an operation to become a success merely by transferring it to Commerce Dept. sponsorship. Rather, opportunity for the establishment and growth of a super-boondoggle is seen in this proposed agency.

The second avowed purpose of the bill is to "hasten the introduction of useful new inventions by affording facilities to independent inventors for evaluation of their inventions and by providing forms of assistance for development of worthy inventions." This is carrying the Commerce Dept. attitude of "let's be kind to the small businessman" to the extreme.

Just how the determination is to be made that the invention is practicable and that there is a reasonable likelihood that its development would lower costs of production or increase industrial efficiency is not stated. Experience has shown that such determination can be made only after the invention is tried out and whether it is to be tried or not depends upon the present condition of the industry in question, what other methods there are of performing the functions which the new invention is supposed to perform, and other pertinent factors. It is felt by many that this is a province of industry rather than of government.

Opportunity is seen here by responsible engineers for every crack-brained inventor who has government backing to set himself up in competition with sound businesses, supported by government funds and equipment. It is felt that political control of engineering research might result, with the government being able to dole out funds and technical assistance to competitors of firms in political disfavor with the administration.

Patents—Last, but far from least, the bill "provides for methods of promoting wider use in private business of the large number of public-controlled patents which have resulted from Government research and development." Here, the link with the Kilgore bill is clear. Patents resulting from federally-financed "basic" research under the Science Foundation would be funneled through the new Commerce Dept. agency, and directed to firms chosen by the agency.

Furthermore, no development is to be undertaken, or advances made, unless the inventor is willing to license the United States Government to use this information without the payment by the United States of royalties or other fees. This is entirely understandable if the United States Government were to use this invention in connection with direct Governmental activities. But it is further provided that the inventor, to be able to take advantage of this provision, must grant non-exclusive licenses to any applicant on a reasonable royalty basis and the "reasonable royalty" is to be determined by the government.

Thus, the same policy which the Kilgore bill would establish in the physical sciences, would be continued and enlarged on in the

commercial application of scientific and engineering research. This legislation is certainly not designed to allay the fears of those who are wary of government intervention into private enterprise. An example of practical scientists and engineers who oppose such intervention can be found in the *Committee Supporting the Bush Report*. As described in the January "Washington Notes" column, these men are opposed to most of the provisions of the Kilgore bill. Further, the use of such old-line agencies as the Bureau of Standards, which has had the cooperation of private industry in the past, in giving "technical service" to the new agency is looked on with some trepidation by engineers who feel that its standards may be lowered thereby.

Wallace's Role—Not content to wait for legislative action, Secretary of Commerce Wallace has obtained executive orders from President Truman, still acting under the War Powers Act, which has enabled him to begin *de facto* organization of the proposed Office of Technical Services. The Office of Production Research and Development has already been transferred to the Commerce Dept. Thus far, its work has consisted mainly of issuing reports of the research accomplished while it was part of W.P.B. Unlike its former functions, however, the Commerce Dept. intends to make of it primarily a "technical service" unit. This is reminiscent of the Smaller War Plants Corp. technical advisory service, which asked for information from large research organizations, in order to transmit this information, gratis, to small businesses who might very well be their competitors.

A further step taken by Wallace has been the transfer of the Smaller War Plants Corp. to the newly-formed Office of Small Business. Thus, the pattern is clear: The legislation of Kilgore and Fulbright is designed to perpetuate in law what Henry Wallace has been able to construct in the name of "emergency."

Fulbright's Activities—Senator J. William Fulbright, recently mentioned as a possible successor to Nicholas Murray Butler as president of Columbia University, seems to be following in the footsteps of Senator Harley M. Kilgore, not only in similarity of ideas but in prolificacy of legislation.

Recently introduced by Sen. Fulbright was a new bill, S.1636, which would amend the Surplus Property Act of 1944, to designate the Department of State as the disposal agency for surplus property outside of the United States and its possessions.

Its second provision, not stated in the opening announcement, is perhaps more important. This section would authorize the Secretary of State to enter into agreements with any foreign government for the use of currencies or credits acquired as a result of surplus property disposal abroad for the purpose of providing for the financing of studies for American citizens in institutions of higher learning located in such foreign countries. The bill would further authorize the use of such funds to provide financial assistance to citizens of such foreign countries in studying in American schools of higher learning located abroad. Finally, the bill would authorize the use of such funds and credits for the purpose of furnishing transportation for citizens of such foreign countries who desire to attend American schools and institutions of higher learning within the United States.

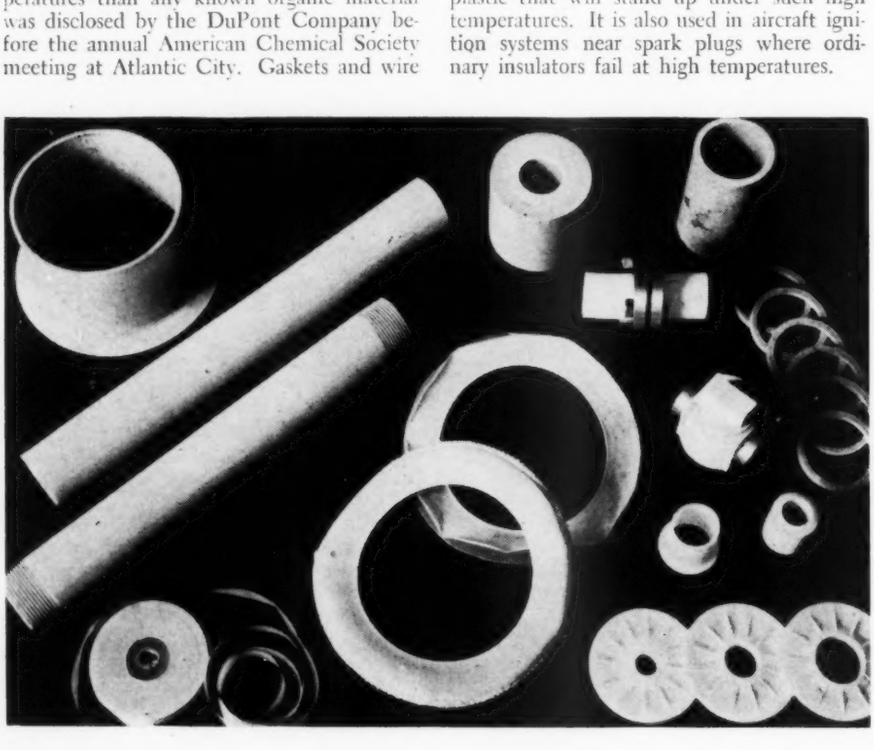
May Fair—Still another recently-introduced Fulbright bill, S.1836, continues the "let's be kind to the small businessman" trend. This bill would establish an annual National Small Industries Fair, to be held for one week during the month of May. To expedite the Fair, an Office of Director of the National Small Industries Fair would be established, with a director at a salary of $10,000 a year, and twelve regional assistant directors.

There being nothing narrow in Mr. Fulbright's makeup, he defines exhibits at the Fair to consist of: "(1) Products in actual production; (2) products not yet in production which the exhibitor wishes to test for buyer response; (3) products which are the property of inventors who are seeking to interest manufacturers."

Mr. Fulbright does not mention an appropriation for a Maypole in this bill.

Acid-Resistant Heat-Stable Plastic

A NEW INDUSTRIAL PLASTIC that withstands acids which dissolve gold and platinum and retains its strength and form at higher temperatures than any known organic material was disclosed by the DuPont Company before the annual American Chemical Society meeting at Atlantic City. Gaskets and wire insulation for jet engines are now made of this plastic, which is unharmed by temperatures up to 575 deg. F. It is the only plastic that will stand up under such high temperatures. It is also used in aircraft ignition systems near spark plugs where ordinary insulators fail at high temperatures.



The plastic material, tetrafluoroethylene resin—to be known by the trademark "Teflon"—was developed in DuPont research laboratories. Limited amounts in the forms of sheets, rods, tubes, coated wire, tape, and fabricated sections are now available for experimental purposes.

Conduits subject to attack by corrosive materials and spacers for coaxial cables conducting extremely high-frequency circuits are other applications. Because its dielectric loss factor is extremely low, even at frequencies up to 3,000 megacyles, it is claimed to be an excellent insulating material for currents of ultra-high frequency.

Aqua regia, chlorosulfonic acid, acetyl chloride, boron trifluoride, hot sulfuric acid, hot nitric acid and boiling solutions of sodium hydroxide do not affect the polymer.

Subjected to a temperature of 572 deg. F. continuously for three months in a recent test, it showed virtually no degradation, and retained all its useful properties. It is not adversely affected by temperatures as low as minus 75 deg. F. Its water absorption rate is practically nil.

Shown in the photograph are some of the numerous applications for "Teflon" tetrafluoroethylene resin: flared tubing, threaded pipe, cylinders, coaxial cable spacers, gaskets, tape, a plug cock, a valve-stem packing and a rod.

Difficulty in fabrication and high cost of manufacture are conditioning factors in the plastic's present stage of development.

RESEARCH NOTES

Plastics Manufacturers Sponsor Research Program

Long-range research in plastics products problems has been started at the Massachusetts Institute of Technology, sponsored by the Plastic Materials Manufacturers Association, made up of the leading producers of plastics.

The work at M.I.T. will be under the supervision of Professor A. G. H. Dietz, of the Department of Building Engineering and Construction, who will act as director of the project. Research will deal with the fundamental engineering properties of plastics and, if necessary, will set up new test methods, similar to those now available for other structural materials.

Included in the M.I.T. research program will be studies of such factors as the effect of molecular structure upon the behavior of plastics; how the rate at which loads are applied affects the strength properties and ductility of these materials; how plastics may be expected to act when loads are applied and removed as in structures subjected to vibration; what the effect of different temperatures may be upon plastics of various types, and what may be the effect of age upon the strength properties.

Existing methods of testing will be examined for their applicability to the plastics, and new methods will be developed, if necessary, to make the needed measurements.

With a measurement of some of the "pure" physical properties by special tests such as are now possible with metals, it is felt that it will be easier to correlate properties of the material and the end use of the product.

It is the aim of the M.I.T. research to correlate physical properties with chemical composition, but none of the research will be aimed at development of new plastics. Only plastics now in existence or which will become commercially available will be studied, and results of such research will be published in scientific journals, with data made available for industrial laboratories of participating companies while the research is in progress.

Representing the plastics association on the steering committee for the project will be Dr. D. S. Frederick, vice president of Rohm and Haas Company, Philadelphia. Assisting Dr. Frederick from P.M.M.A. will be Harold W. Paine of E. I. du Pont de Nemours and Company, Inc., John H. Adams of Bakelite Corporation, W. C. Goggin of the Dow Chemical Company, Howard J. Nason of Monsanto Chemical Company, and L. W. A. Meyer of Tennessee Eastman Corporation.

Associated with Dr. Dietz on the M.I.T. Plastics Committee are Dr. E. A. Hauser, Professor E. R. Schwarz, Dr. W. H. Stockmayer, and Dr. W. M. Murray. The association has named a test specimen committee composed of L. A. Sontag of Durez Plastics and Chemicals, Inc., W. E. Gloor of Hercules Powder Company, and J. A. Murray of Plaskon Division, Libbey-Owens-Ford Glass Company.

Dr. Robert Burns of Bell Telephone Laboratories will act as representative of the American Society for Testing Materials in an advisory capacity to the steering committee. As dean of engineering at M.I.T., Professor Edward L. Moreland will have overall direction of the project.

Gas Turbine Laboratory To Be Established at M.I.T.

Gifts totalling half a million dollars for establishing a gas turbine laboratory at the Massachusetts Institute of Technology for graduate instruction and fundamental research in this new field of engineering, were announced by Dr. Jerome C. Hunsaker, head of the department of aeronautical and mechanical engineering.

Recent advances in gas turbine design indicate significant possibilities for the future and emphasize the need for specialized technical studies and the discovery of new techniques required in the construction of machines operating at extremely high speed, high temperature and high compression. The Institute's new laboratory, which is to be completed as soon as possible, will include a supersonic wind tunnel and facilities for research on the elements of compressors, combustion devices, jets, and gas turbines. Test facilities will also be provided for the operation under controlled conditions of such devices.

Early realization of the project is made possible by substantial grants from a group of leading industries, all of which will benefit from fundamental research on gas turbines. Those contributing to the program include: Alfred P. Sloan, Jr., chairman of the board of General Motors Corp.; the General Electric Co.; the Westinghouse Electric Co.; the United Aircraft Co.; the Hooven, Owens, Rentschler Co.; and the Curtiss-Wright Corp. Professor Edward S. Taylor is project engineer for the laboratory.

Westinghouse Opens Western High Frequency Laboratory

A new laboratory to which western industry may bring for study its problems of speeding production by making use of high frequency heating, was opened by the Westinghouse Electric Corporation in Los Angeles. In charge of the laboratory is Dr. Russell A. Nielsen, formerly of the company's Research Laboratories at East Pittsburgh, Pennsylvania, assisted by a staff of engineers.

Included in the laboratory's equipment are many dielectric heating units: The five kilowatt dielectric heating generator in the laboratory operates at a frequency of either five or 30 megacyles—5,000,000 or 30,000,000 cycles. The 20-kilowatt induction heating generator operates at a frequency of 450 kilocycles, well below the familiar commercial broadcasting band. On a one-kilowatt generator, Dr. Nielsen demonstrated the high speed setting of glue, a process which is materially speeding the fabrication of plywoods and enabling the manufacture of thicker, stronger sections than were possible by earlier processes. Thermosetting glue in the demonstration was set in less than one minute.

Dr. Russell A. Nielsen, in charge of the High Frequency Laboratory opened by Westinghouse in Los Angeles, changes the power level on a transformer in a 20-kilowatt 450-kilocycle industrial radio frequency generator used in industrial induction heating.

Naval Ordnance Award Presented to Industrial Designer

The Naval Ordnance Development Award was presented to the staff of Walter Dorwin Teague by Captain J. S. Champlin, Bureau of Ordnance, U. S. Navy, Washington, D. C. "for distinguished service to the research and development of Naval Ordnance," in a dinner given in New York on March 22.

In addition to the staff award, Certificates for Exceptional Service to Naval Ordnance Development were presented to 18 members of the staff.

Commander C. E. Haugen, USNR, presented a similar award to the Los Angeles staff of the Walter Dorwin Teague organization at a dinner Tuesday night, March 26th, at the Hotel Huntington, Pasadena, California. Dr. L. A. Richards of California Institute of Technology, who is largely responsible for the development of Naval rocket launchers, delivered an address. The Los Angeles staff was originally established at Pasadena in 1943 for collaboration with Cal Tech in the development of the Naval rocket program.

American and Foreign Engineer Equipment Demonstrated

U. S. Army Engineer equipment and its counterpart in captured German and Japanese equipment was demonstrated recently at the Engineer Board, Fort Belvoir, Virginia. The purpose of the exhibit was to enable engineers and manufacturers to inspect existing U. S. and foreign equipment, with a view to further development.

The exhibit included more than 200 separate items of engineer equipment, classified as follows: (1) Survey, mapping, and photogrammetric equipment, including a complete display of Carl Zeiss instruments and featuring the Stereoplanigraph; (2) Electrical and electronic equipment with the latest American and foreign developments in mine detection, the odograph and pedograph, and infra-red; (3) Construction equipment —comparative studies and demonstrations of American and foreign design in earth movers, augurs, trailers, road and airfield equipment, and tractor-mounted cranes; (4) Water supply equipment and distillation units, and airborne well-drilling machines; (5) Bridging equipment, storm and assault boats, power launches, pontons, outboard motors; (6) Demolition devices, including underwater instruments.

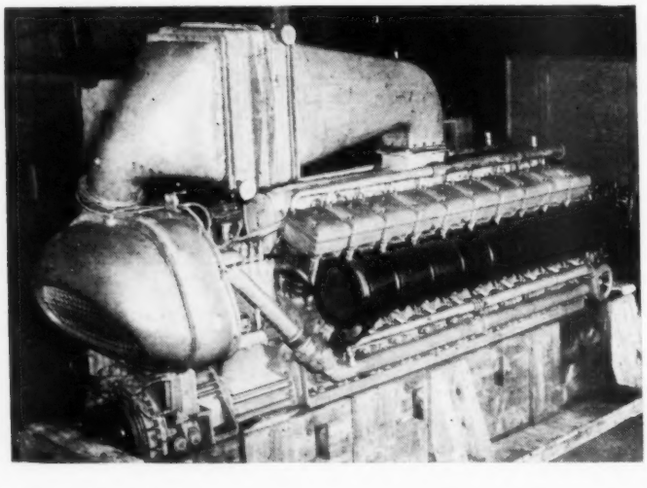
Among items of interest to American science and industry were German, Italian, and Japanese crawler-type tractors and air and oxygen compressor equipment; a Japanese 250 kg. internal combustion pile driver; a Japanese oscillating hand water pump; the Japanese "Ishii" and German "Katadyn" water filters; a German ponton "B" bridge of stainless steel; a German block and tackle, with plastic block and silk or rayon rope; a Japanese 2 cylinder "Kinuta" gasoline outboard motor; remote indicating compasses and an Italian vehicular magnetic compass; capsules developed by the Germans for preventing fungus fouling of optical instruments; and a German one-piece amphibious suit with inflation charger.

Shown in the photograph is the low ground pressure tractor with bulldozer, which was designed to furnish a crawler tractor with ground pressure sufficiently low to permit construction dozer work and the towing of equipment, in mud areas, swamps and jungles. Developed primarily for Engineer construction battalions by the Caterpillar Company and International Harvester Company, the low ground pressure tractor dozer is of value where ground support is insufficient for crawler tractors of standard design. Pilot models are under test, and satisfactory results are being obtained. These are modifications of the standard Corps of Engineers medium crawler tractor, having a wide gage extended tractor frame equipped with 32 in. wide track shoes in lieu of the standard 20 in. shoes.

The gross weight and dimensions of these tractors, equipped with bulldozers, does not exceed the capacity of the standard Engineer truck trailers when rapid transportation is necessary. The bulldozer blade is mounted inside the tracks rather than by use of the conventional mounting arms, which are outside of the tracks. The modification of the design permits the use of other standard tractor attachments such as A-frame cranes, revolving cranes and winches.

Army Displays German Transportation Equipment

Captured German diesel engines, locomotives, railroad cars, a minesweeper, and other German transportation equipment was recently inspected by representatives of American science and industry at Fort Monroe, Virginia. The exhibit, which included more than 35 items, was prepared by the Army's Transportation Corps Board. The equipment was obtained in Germany by the Transportation Corps Technical Intelligence Team, under orders from the Office of the Chief of Transportation.



Various Diesel engines, ranging from a 45-horsepower engine to a 2,500-horsepower engine which weighs 10,000 pounds, were on display. These engines were developed by the Germans for marine use and for locomotives. The purpose of exhibiting the engines was to permit investigation by American engineers of the German designers' basic technical developments, with a view to commercial adaptation and mass production.

A 135-foot Diesel rudderless minesweeper included in the exhibit has vertical type (cycloidal) propellers and is the first craft of this type to cross the Atlantic under its own power.

Other items on display were a supercharged, 2,000-horsepower submarine engine; a high-speed locomotive with individual V-type engines; a 17,000-gallon tank car, constructed so as to require no underframe; a Class 52 condensing locomotive; a lightweight Diesel locomotive with fluid mechanical drive; 12-cylinder, V-type, air-cooled Diesel Engines; gas turbine driven superchargers (in three sizes); three electrical speed transmissions; several Voith-Schneider cycloidal propeller units, including control stand and accessories; and a propeller with welded, hollow stainless steel blades.

Shown in the photograph is the V-20 Marcedes Benz diesel engine, which develops 2,500 hp. at 1,630 mph. The total cylinder capacity is divided into many individual cylinders, which are of welded steel. The crankcase is made of aluminum alloy and reaches to the upper part of the cylinder. The supercharger is gear driven from the aft end of the crankshaft. This type of diesel engine was used on German "E" boats, in groups of two or three.

Programs of Society Meetings

THE PITTSBURGH SECTION of the Electrochemical Society is sponsoring a Symposium on "Surface Reactions" at the Mellon Institute of Industrial Research on June 7.

The following program has been arranged:

"The Chemical Structure of Thin Oxide Films on Metals and Alloys Formed at Elevated Temperatures," Dr. J. W. Hickman, Westinghouse Research Laboratories.

"Surface Area Measurement on Metal Surfaces," Dr. D. H. Emmett, Mellon Institute of Industrial Research.

"Theory of Oxidation and Tarnishing," Dr. W. E. Campbell, Bell Telephone Labs.

"Relation Between Solid Binding and Surface Structures," Dr. Frederick Seitz, Carnegie Institute of Technology.

"Reactions Attendant with Carburization-Decarburization of Steels," Mr. J. K. Stanley, Westinghouse Research Labs.

"Thermodynamics of Surfaces and Solutions," Dr. J. C. Warner, Carnegie Institute of Technology.

* * *

Several interesting technical symposiums are being prepared for the 1946 Annual Meeting of the American Society for Testing Materials to be held at the Hotel Statler in Buffalo during the week of June 24. Also under way will be the Seventh A.S.T.M. Exhibit of Testing Apparatus and Related Equipment and the Annual Photographic Exhibit. In addition to the some fifteen to twenty technical sessions necessary for the program there will be a large number of meetings is the Society's technical committees.

Two of the symposiums which should be of interest will involve Testing of Parts and Assemblies, the second covering Fatigue of Materials. The Society for Experimental Stress Analysis is joining with A.S.T.M. in sponsoring the symposium on Testing of Parts and Assemblies.

Further topics will include round table discussion on Test Methods of Freezing and Thawing Concrete to Determine Durability, a session dealing with Tools for Identification of Water Formed Deposits, and a Symposium on pH Measurement.

The Annual Edgar Marburg Lecture, always an outstanding feature of the meeting, will be presented this year by Dr. J. J. Mattiello, Technical Director of the Hilo Varnish on the subject "Protective Organic Coatings as Engineering Materials."

* * *

First technical session of the newly formed Machine Design group of the American Society of Mechanical Engineers will be held on the afternoon of Monday, June 17, 1946, during the semiannual meeting of the Society in the Hotel Statler, Detroit. All engineers interested—whether A. S. M. E. members or not—are invited to attend. Following is the program for the session:

"Application of Tables for Helical Spring Design" by H. F. Ross, United Shoe Machinery Corp., Beverly, Mass.

"High-Speed Chain Developments" by A. W. Meyer, Brown & Sharpe Mfg. Co., Providence, R. I.

"Evaluating Self-Locking Nuts for Industrial Equipment" by Wm. E. Horenburger, Elastic Stop Nut Co. of America, Union, N. J.

MEETINGS

Mid-America Exposition—Industrial exposition May 23-June 2, Public Hall, Cleveland 14, Ohio.

American Foundrymen's Association – Annual meeting, including symposium for design engineers on "Engineering of Gray Cast Iron," May 6-10, Public Auditorium, Cleveland.

Metal Powder Association—Spring meeting, June 13, Waldorf-Astoria Hotel, New York.

American Gear Manufacturers Association—Annual meeting, June 3-5, The Homestead, Hot Springs, Virginia.

American Society for Testing Materials—Annual meeting, June 24-28, Hotel Statler, Buffalo, N. Y.

Society for Experimental Stress Analysis—Spring meeting, including symposium on fatigue of manufactured parts, held jointly with A.S.T.M., June 24-26, Hotel Statler, Buffalo, N. Y.

American Electroplaters' Society – National meeting, June 17-20, William Penn Hotel, Pittsburgh, Pa.

Colloid Symposium – Annual symposium, May 28-29, University of Wisconsin, Madison, Wisconsin.

National Association of Corrosion Engineers –Annual meeting, May 7-9, President Hotel, Kansas City.

American Institute of Electrical Engineers—Summer Convention, June 24-28, Hotel Statler, Detroit.

Society of Automotive Engineers—Summer meeting, June 3-7, French Lick Springs, French Lick, Indiana.

Acoustical Society of America—Annual meeting, May 10-11, Hotel Pennsylvania, New York.

American Society of Mechanical Engineers –Semi-annual meeting, including first technical session of the Machine Design group, June 17-20, Hotel Statler, Detroit. Aviation Division meeting, June 3-6, University of California, Los Angeles. Power Division meeting, June 12-15, Hotel Schroeder, Milwaukee. Applied Mechanics Division meeting, June 21-22, Hotel Sheraton, Buffalo.

Engineering Societies to Help Restore Technical Libraries

A NATION-WIDE CAMPAIGN looking toward restoration of engineering libraries in war-devastated areas overseas was announced recently by the American Society of Mechanical Engineers, through its Committee on International Relations. An appeal was issued to engineers throughout the country to assist with gifts of technical books and periodicals to replace those lost or destroyed. Donations of money will be used to buy new books for foreign technical libraries.

Supporting the movement besides the A.S.M.E. are four other national Engineering Societies: American Society of Civil Engineers, American Institute of Mining and Metallurgical Engineers, American Institute of Electrical Engineers, and American Institute of Chemical Engineers, all represented on the committee. The aggregate membership of the five societies is more than 80,000. Headquarters of the committee is in the Engineering Societies Building, 29 West 39th Street, New York.

The Committee on International Relations has met with an enthusiastic response from foreign officials, engineers, librarans and educators with whom it has been in contact, according to Chairman Robert M. Gates of New York, a past president of the A.S.M.E. Requests have come from China and the Philippine Islands, and for a specific list of books needed in Czchoslovakia. The goal of the campaign is to begin restocking libraries despoiled by war, to modernize those long shut off from the outside world, and to advance technical progress generally, with mutual benefits to engineers both at home and abroad.

DISCUSSIONS AND COMMENTS FROM READERS

Component Design and Performance Specifications

To the Editor:

Col. T. B. Holliday's reply to Mr. H. S. Spencer's comments (page 865, December 1945 issue) on the Colonel's article "Failures of Electrical Equipment from Poor Mechanical Design" justified further discussion. Satisfactory field performance of any component, for aircraft or otherwise, depends on engineering effort rightly applied in the branches of material choice, component design, and component application, and performance in no one of these branches fully accounts for field failure. Our experience indicates the major cause of field trouble, if selection should be attempted, has been inadequate application engineering. This has, in some instances, resulted from failure of the component manufacturer to make available product information needed for correct application. Much more often, however,

misapplication resulted although product information was available.

Delay in bringing back to the component manufacturers information and samples illustrating field troubles slowed corrective action in some cases. Additionally, corrective measures developed by component manufacturers as answers to field troubles were, in our experience, slow in reaching the field. As a result, such troubles continued even after they had been recognized at service headquarters and remedies made available.

Complete sealing of a switch or any other electrical device should not be advanced as the only answer to field requirements, nor as the solution for every failure already experienced. Test procedures must first be developed that will dependably establish component performance under carefully defined conditions (including environmental). These conditions should represent, in the greatest possible degree of accuracy, field operating requirements that components will be required to meet. Then the component manufacturer should be allowed any degree of engineering freedom he chooses to make use of, so long as his product meets such test requirements, plus such definition of envelope, terminal arrangement, and so on, which may be necessary to accomplish field interchangeability. If our combat equipment of the future is developed on a basis of "It must be made so" rather than "It must perform so" the component manufacturer, the component user, and the Armed Services will all suffer. R. A. MILLERMASTER

Assistant Mgr. Development Department
Cutler-Hammer, Inc.

Wants Hardened Walls To Prevent Cylinder Wear

To the Editor:

We make a certain type of cylinder for circular knitting machines. It is about 4 in. long, with outside diameter of about 3½ in. and an inside diameter of about 3 in. We mill slots lengthwise on the outside of the cylinder for the needles to work in, these slots being about $\frac{5}{32}$ in. deep, and it is necessary that the walls of these slots be hardened to prevent excessive wear by the needles.

We have much difficulty in hardening these walls properly, as the cylinder tends to warp when heat treated. There is no necessity for the whole cylinder to be hardened, though there is no objection to it, provided the walls are hardened. We would like some advice on the best type of steel to be used for this application and the best way to harden the walls of the slots without warping the entire cylinder.—J. J. A.

"The Mind To Work Is the Root of Progress"

To the Editor:

Your March editorial strikes me as being a very sound piece of philosophy. I am sure that it would be an excellent thing for all of us if such counsel were taken to heart by more people throughout the land. It seems to me that you have made a definite contribution to our present day sociology in this editorial and I only hope that it is read carefully and considered seriously by all of your readers. —W. J. KING

General Electric Co.

To the Editor:

The editorial in the March issue of PRODUCT ENGINEERING is a splendid presentation of the problem before us today. I think you have done the public a service in placing emphasis on the fact that a new, or should I say old-fashioned, concept of life must be instilled in a great portion of our population.

In our engineering departments that we have under our control we can attempt the creation of better moral standards as you suggest. I have always felt that the subject of personnel was the most important phase of my job. —DON SMELLIE

Chief Engineer
The Hoover Company

To the Editor:

I liked your March editorial, not because I agree with it, but because it touches on ground that interests me intensely. I have much more faith in the essential soundness of people than to believe that all but a few prefer to work for the return rather than in response to an urge which they do not attempt to analyze and which they do not have to control.

My experience in the handling of people has been almost entirely limited to technical personnel, in whom there always seems to be "the mind to work."

As an engineer myself (it is inconceivable to me that anybody could stand the moral strain of working for a living in any pursuit other than what he prefers. I have noticed recently (in my experience) a new phenomenon: there is a general drift of engineers, laboratory assistants, draftsmen and other personnel away from corporations. These men do not leave in order to do similar work for some other concern; they leave us in order to engage in some entirely different pursuit; engineers have left us in order to operate a dude ranch, run a package store, go into partnership in a fish market, operate a line of cabins, to run a flying school and barnstorming activity and many other types of business. What I find most surprising is that so many of these men are relinquishing their technical work for straight business.

I have found that exactly the same drift is taking place in many manufacturing companies and, on further discussion of this phenomenon with members of firms so affected, I have come to the conclusion that it is not a symptom of demoralization at all. I believe that the urge which causes all these men to pull out of companies which they have been happy to serve for many years derives from their realization that employment by a large organization is no longer any guarantee of security. Many men would prefer a conservative income rather than an opportunity to make much more money which would be offset by comparative lack of security.

So far as my own observation goes, any want of "the mind to work" among technical personnel is something less than 1 percent of the people with whom I work. Our worst lack seems to be enlightened leadership, and the trouble starts at the top.

—B. M.

To the Editor:

I have read your editorial very carefully and I certainly agree with everything that you have said in it. It would be a very good thing if everyone would read this and apply to themselves all of the truths which you have outlined. We see every day conditions arising which make the things that you have said stand out vividly in our own mind. I do not know what can be done to offset some of these things, but you have pointed out many things which might be done and I believe if these suggestions were followed, we would get better conditions.

—E. D. VANCIL.

Manager of Engineering,
The Cincinnati Milling Machine Co.

Comment on Patent Suppression Abstract

To the Editor:

I have read the article "The Suppression of Patents" by Alexander Morrow, abstracted in the February issue of Product Engineering. Those of us who have practical experience with these matters know that it is very rarely that there is any suppression of patents. Anything that is profitable among patents is rarely left unused. But thousands of patents must be taken out before one of them is sufficiently valuable to constitute the basis for a commercial operation. There is no company in the United States rich enough to put into commercial operation all of the patents that it takes out.

The error in Mr. Morrow's thinking is this: the primary purpose of patents is not use. The primary purpose is publication of inventions through patents and the reward of a monopoly is for publication. Our Constitution recognizes this fact and so have our statutes by not requiring the use of a patent, but only its publication to get a reward. The value of publication is enormous in our economy because it aids thousands of inventors and companies to profit from it in their own research and development engineering work even though it might not be practical enough or commercial enough to justify its owner to go into business under it.

The whole subject of suppression of patents is in difficulty for this lack of understanding of the practical reason why our founding fathers and our great legislators have always insisted that the patent reward is for publication and not for manufacture. Mere manufacture would only benefit a company and a limited number of its customers, but publication benefits everyone interested in the entire nation.

Mr. Morrow has missed the point in his article. —H. A. Toulmin, Jr.

Standard Small Motors Can Fit Most Design Needs—II

CYRIL G. VEINOTT
Section Engineer, Small Motor Division, Westinghouse Electric Corporation

Fundamental principles of operation, proper application, and performance of shaded-pole, synchronous, universal, direct current, and other types of fractional horsepower motors are discussed.

IN ADDITION to the split phase and capacitor types of motors discussed in Part I of this article in PRODUCT ENGINEERING, April 1946, page 313, there are other diversified types of standard small motors that often fit the needs of product designers. These types include shaded-pole, synchronous, universal, direct current, and others.

SHADED-POLE MOTORS. Shaded-pole motors are built in ratings up to about 1/20 hp. The rotor is of the conventional squirrel-cage type and the stator consists of salient poles. A portion of each pole is "shaded", that is, surrounded by a short-circuited coil. Without shading coils, when excited by single phase a.c. this motor tends to set up a magnetic field, pulsating in magnitude, but stationary in space. Adding shading coils has the effect of reducing the flux within each shading coil and causing it to lag in time phase behind the flux in the unshaded portion of the main pole. Consequently, the effect is to produce a shifting field, the field shifting from the unshaded to the shaded portion. This shading coil stays in the circuit at all speeds and the motor requires no starting switch. Absence of starting switch, commutator, brushes, or any sliding contact whatever, makes this a reliable and sturdy motor.

The typical performance curve of a shaded-pole motor in Fig. 1 reveals that the locked-rotor power input is only about one-third more than that for full load. Because of this, many shaded-pole motors can be stalled indefinitely without harm, thus the shaded-pole motor is especially suitable for many applications.

Shaded-pole motors are well suited to fan applications where speed control is required. When used with a fan load, speed control can be obtained by changing the voltage impressed on the motor by: A series choke, a tapped winding, or an autotransformer. By any of these three means, speed is changed by changing the slip. Therefore this method has the same inherent limitations as discussed for the single-value capacitor motor.

Shaded-pole motors are used for driving all types of control and regulators, animated signs, or almost any application requiring a very small amount of power. The ordinary type of shaded-pole motor cannot be reversed. However, with two shading windings, and by closing either one or the other, either direction of rotation can be obtained.

One interesting application of a reversible shaded-pole motor is to turn the micrometer screw up and down in an electronic micrometer. When the micrometer screw makes contact the motor is reversed by means of an electronic circuit; when the contact is broken the same electronic circuit again reverses the motor, thereby causing a hunting action that places the micrometer screw just barely in contact with the object it is measuring. An advantage of this automatic electronic micrometer is that it can be used to take rapid and exact measurements of elastic bodies such as rubber, or of elements like various diaphragms used in aircraft instruments while they are being tested inside an altitude chamber. By proper adjustment of the motor speed, the range of motion of the micrometer tip can be limited to perhaps less than ten millionths of an inch. The micrometer will follow with this accuracy any changes in the position of a diaphragm, for example, as the pressure or temperature on it is changed.

SYNCHRONOUS MOTORS. Small synchronous motors are widely used in instrument

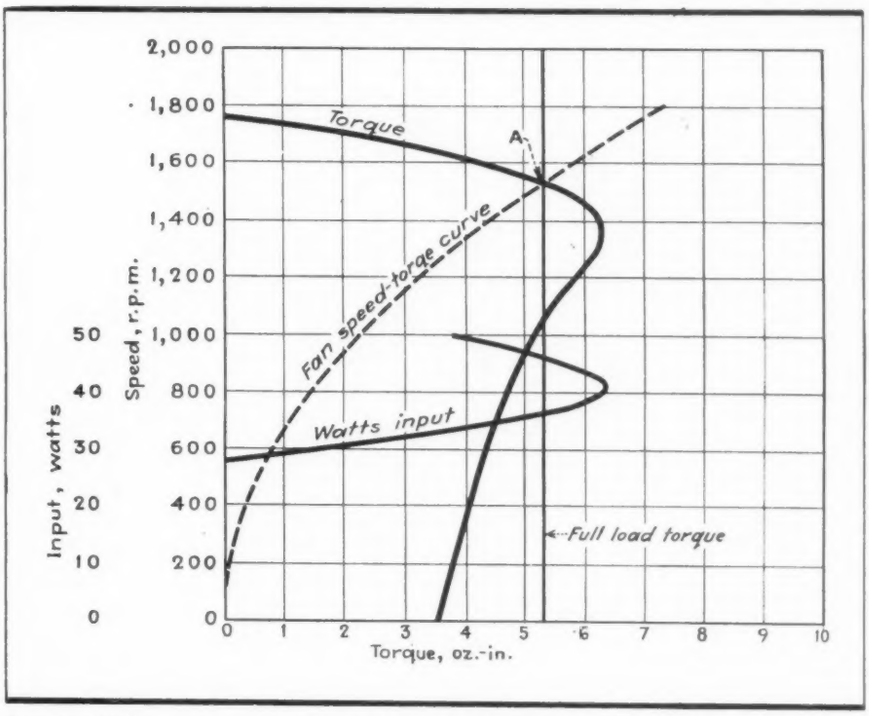


Fig. 1—Performance curves for a shaded-pole motor. To obtain best efficiency the motor is worked near the pull-out torque where the fan speed-torque curve crosses at *A*. Because the watts input for locked-rotor conditions is only 50 watts as compared to 37 watts for full load, stalling the motor does no harm.

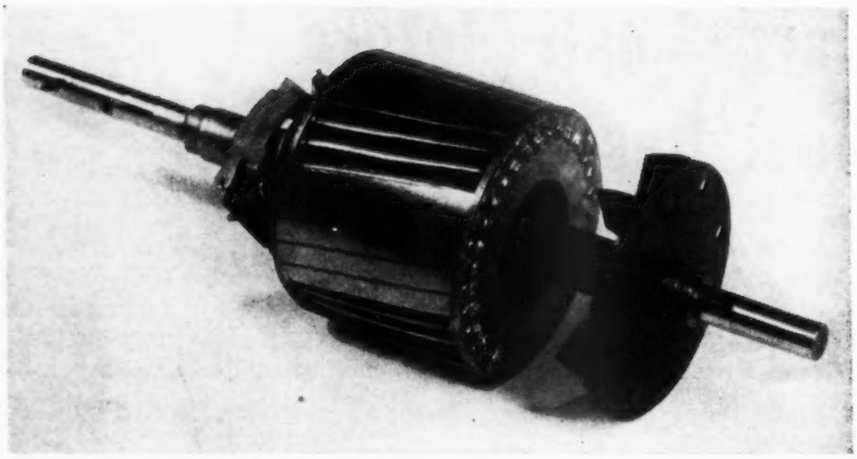


Fig. 2—Rotor of 4 pole reluctance-type synchronous motor. There are as many notches in the rotor as there are poles. The squirrel-cage bars improve the starting characteristics.

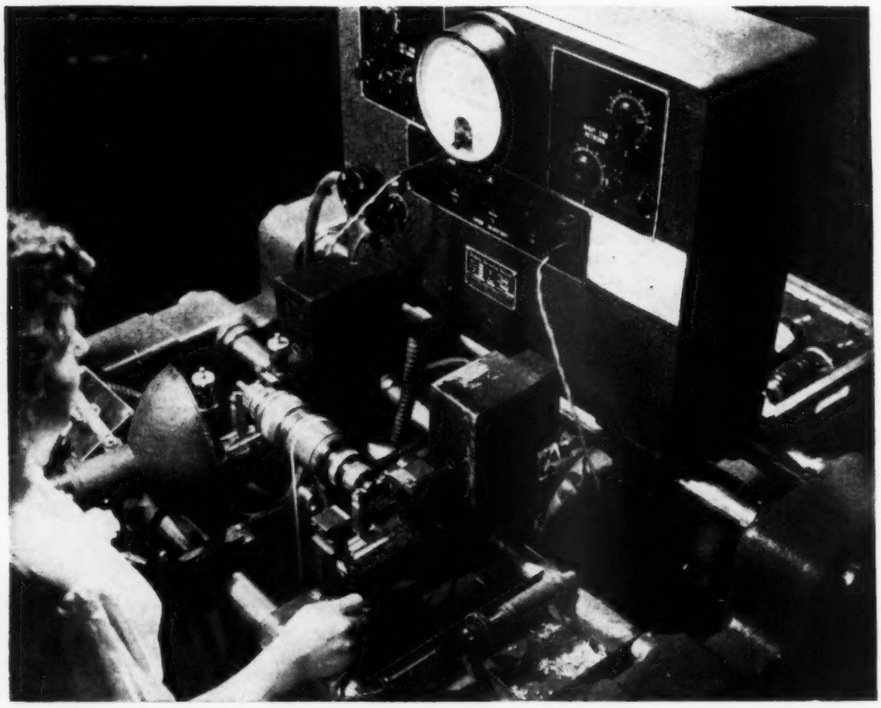


Fig. 3—A reluctance-type synchronous motor spins this four commutator "Flying Tiger" dynamotor armature at absolutely constant speed while it is being balanced by this electronic balancing machine.

and control work where exact speed is necessary. Unlike large synchronous motors using direct current excitation, small synchronous motors such as used in instruments are non-excited. There are numerous ways of constructing small synchronous motors, most of which can be grouped as either the reluctance type or the hysteresis type.

Reluctance type motors generally have rotors that look like the conventional squirrel-cage rotor used in induction motors, except that there are grooves, or notches, cut in the iron as shown in Fig. 2. If the stator is wound for conventional polyphase excitation, a rotating magnetic field will be formed, and the projections caused by the notching of the rotor will lock in with this field and will rotate at the same speed as the magnetic field itself. There are as many notches in the rotor as there are poles. As in Fig. 2, squirrel-cage bars are left in the cut-out portion to provide better starting characteristics.

Reluctance type synchronous motors operate satisfactorily with a single phase supply and with any of the popular starting arrangements. For example, the reluctance type motor can be wound as a split-phase, capacitor-start, or single-value capacitor motor. Reluctance type synchronous motors are generally larger than the corresponding induction motors. For any given set of parts, only about one-third to one-half of the power can be obtained from the synchronous motor as can be obtained from an induction motor. That is, a 1/12 hp. synchronous motor would be about the size of a corresponding 1/4 hp. induction motor.

An important consideration when applying synchronous motors is that the inertia of the connected load must be kept low because the rotor poles must lock in suddenly with the rotating field. This cannot be done if the connected inertia is too high.

Two common applications of reluctance synchronous motors are for driving recording and temperature indicators, and for aircraft instrument applications where 400 cycle polyphase synchronous motors provide precise speed. Another interesting application of synchronous motors is to drive a dynetric balancing machine, such as in Fig. 3. This is an electronic balancing machine in which constancy of the speed of the apparatus being balanced is necessary for precise determination of the unbalance. For this reason, a split-phase reluctance type synchronous motor is used. Other interesting applications of reluctance type synchronous motors include facsimile picture transmitters and receivers, motion-picture projectors, polar oscillographs, phonograph recorders and testers for calibrating centrifugally operated fuses.

Hysteresis type synchronous motors are also widely used. They are not, as sometimes supposed, merely small editions of large salient pole synchronous motors using permanent magnets instead of d.c. excitation. Hysteresis motors have no salient poles, but merely smooth rotors.

Fundamental principles of operation can perhaps be best understood by considering a conventional polyphase induction motor with a cylindrical rotor and no squirrel-cage winding. With the primary excited and the rotor at standstill, there are iron losses in the rotor consisting of hysteresis and eddy-current losses. The hysteresis loss in the rotor expressed in watts is equal to the hysteresis motor torque expressed in synchronous watts.

For example, suppose that the hysteresis loss in the rotor at standstill is 10 watts; then the standstill torque developed by the motor is the torque required to deliver an output of 10 watts at the synchronous speed of the motor. Theoretically, with a polyphase synchronous motor of the hysteresis type, the hysteresis torque is the same at all speeds from standstill up to synchronous. But practically, there are deviations. Single phase hysteresis motors may have a stator winding of any of the following kinds: Split-phase, capacitor-start, single-value capacitor, or shaded-pole.

A hysteresis motor with the rotating secondary surrounding the stationary primary

is generally called an umbrella type. The smooth rings inside the rotor are made of a material having a high hysteresis loss per cycle. Such a motor-driven gyroscope is shown in Fig. 4 and it has the advantage of unvarying speed at all altitudes. Shaded-pole hysteresis type synchronous motors are widely used in meters, recording instruments, clocks, and timing devices. One design operates from a 60 cycle circuit at a speed of 600 r.p.m. and requires but 1.6 watts input with five-millionths of a hp. output.

POSITION INDICATORS. Certain types of small motors with three phase wound rotors are used as a position indicating system between remote points.

A simple system is shown in Fig. 5. It consists of two motors interconnected electrically but physically separated from each other. When electric power is applied, the rotors of the motors do not run but remain stationary. However, when one rotor is turned mechanically, the rotor in the other motor rotates precisely the same number of turns or part of a turn. Such a system can, for example, transmit the exact level of liquid in a tank located in the yard to a remotely located office. A float in the tank mechanically turns the rotor of the "transmitter" motor as the level of the liquid changes. Inside the office, the "receiver" motor makes exactly the same number of revolutions and the tank level is read directly from the dial connected to the receiver motor.

As shown in Fig. 5, each unit of the sending and receiving end has a single-phase primary, usually of salient-pole construction. Each secondary uses three-phase windings. It is not particularly important whether the primary or the secondary is the rotating member, and position indicators are built both ways.

The basic principle of operation is fairly simple. When power is applied to the two primaries, the secondaries assume "corresponding positions." If the machines are of identical design, the two secondary voltages in the respective phases are equal and opposite in magnitude. The voltages in all three of the windings are in time phase because they are all induced by the same single phase source and there is no rotating field.

If the rotor of the transmitter is turned a little, the voltages induced in the respective three windings of this machine are no longer the same as the voltages induced in the respective three windings of the receiver. Consequently, since the voltages are no longer equal and opposite, current circulates between the two secondary windings. The circulating current develops a torque in both machines. The torque developed in the transmitter resists the mechanical motion turning the rotor; the torque in the receiver tries to move the rotor to a position where the secondary voltages are again equal and opposite. Thus, the receiver can follow the motion of the transmitter, and if not restrained, it does so with a high degree of accuracy.

A position indicating system may have more than one receiver. Such systems are widely used in industrial work for remote indication of the position of various meters, valves, and devices. Unfortunately, there is no generally accepted term for a position indicator. They are known by various names such as, position motor, selsyn, autosyn, diehlsyn, and synchrotie. For specialized applications, such as remote indicating aircraft instruments, other systems such as magnesyns and the d.c. type of position indicators are often used.

UNIVERSAL MOTORS. Universal motors are essentially d.c. series motors, except that the field poles are laminated. Universal motors will operate from d.c. or a.c. of frequencies up to 60 cycles. Ratings up to 1/4 hp. are usually salient-pole construction.

Speed-torque characteristics of a typical universal motor are shown in Fig. 6. For any given load, the motor runs slower on 60 cycles than it does on d.c. or on 25 cycles. The two main reasons why the performance is different on a.c. than on d.c. are reactance voltage and saturation effect. Reactance voltage tends to make the motor run slower on a.c. and saturation effect causes it to run faster than on d.c. power. Reactance voltage is the voltage drop caused by the reactance of the motor, a voltage drop that is not present in d.c. operation. Therefore, the voltage available for the armature is less on a.c. operation than on d.c. operation, and consequently the motor runs slower. However, because of the effect of saturation, a given root mean square value of a.c. will produce less r.m.s. flux than the same d.c. power. Therefore, the net result of the saturation effect is to reduce the flux in the motor, and thus increase the operating speed at any given current. Since the curves for d.c. and 25 cycle operation in Fig. 6 are close together over the operating range, it is evident that these two effects just about cancel each other at a frequency of 25 cycles and excellent "universal" operation is obtained. However, at 60 cycles the reactance effect predominates, causing a slower speed on 60 cycles. This is generally characteristic of the salient pole type.

Better universal motor characteristics are produced by compensated universal motors. Compensating windings neutralize the flux that would be set up by armature reaction and prevent field distortion. Neutralization of this flux also reduces the reactance voltage. The overall result is the improved characteristics of Fig. 7. Applications requiring more than 1/4 hp. generally use compensated universal motors.

Split-series universal motors are particularly useful for applications requiring reversing service or speed control. The motor has two field windings, one for each direction

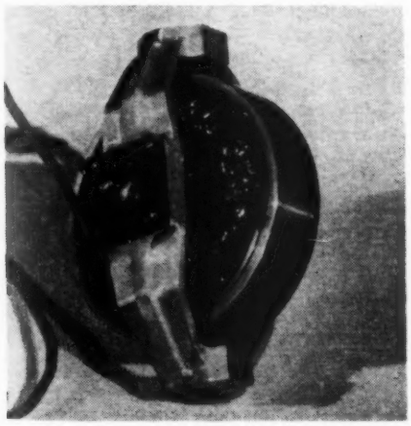


Fig. 4—This small instrument gyroscope is driven by a polyphase hysteresis synchronous motor. Unit is interchangeable with an air-driven gyroscope.

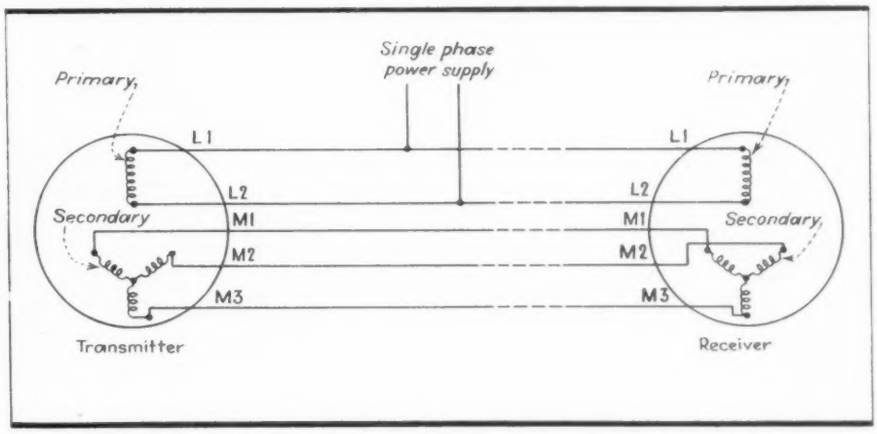


Fig. 5—Schematic diagram of two position indicators with single phase primaries. When the primaries are excited neither motor runs. However, when the rotor of the transmitter is turned mechanically, the rotor in the receiver will turn and follow exactly the movement of the transmitter.

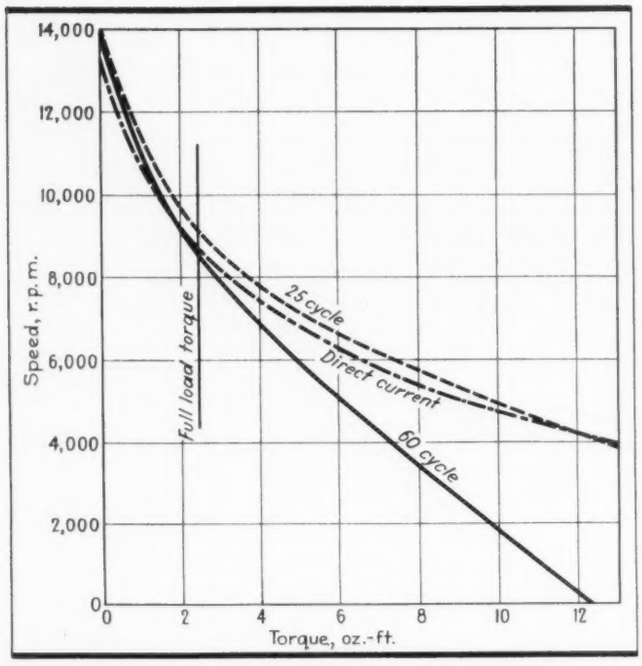


Fig. 6—Speed-torque curves of a 1/4 hp., 8,000 r.p.m., salient-pole noncompensated universal motor when operated on d.c., 25 cycle and 60 cycle alternating current.

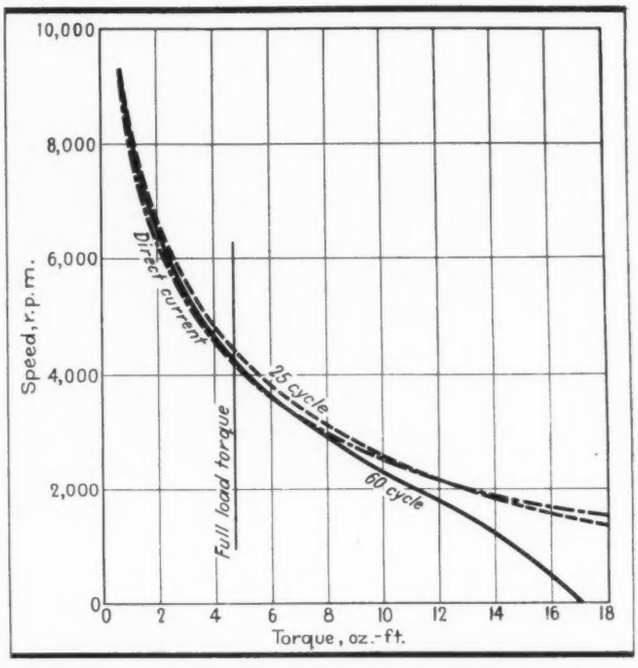


Fig. 7—Speed-torque curves of a 1/4 hp., 4,000 r.p.m., compensated universal motor when operated on d.c., 25 cycle, and 60 cycle, power showing improved operation.

of rotation. Simplicity of reversing and ease of control are the same as that for the single-value reversible capacitor motor previously discussed. Universal motors with tapped fields are often used for speed control. However, they must be carefully applied because of the inherent varying-speed characteristic of universal motors.

Relatively constant speed can be obtained from a universal motor by the use of a centrifugally operated governor, connected as in Fig. 8. When the speed of the motor increases above a pre-set value, centrifugal force opens the contacts. The contacts do not open the motor circuit but merely insert resistance in series with it.

Some governors are built for only single speed of operation. Others are adjustable for a range of speed, the adjustment being made with the motor either stopped or running, depending upon the design. For example, a governor controlled universal motor is used on an electric typewriter and the governor speed setting is changed by a simple lever. Changing the speed of the motors changes the force of the blow of the type bar, and consequently the number of carbon copies that can be typed.

Universal motors are widely used on motion picture projectors, particularly for the 16-mm. size. In some applications, two speed governor controlled motors are used; one speed is for sound pictures at 24 frames per second, and the other for silent pictures at 16 frames per second. The two speeds are obtained by the use of two separate governors; a simple switch selects the speed by selecting the proper governor.

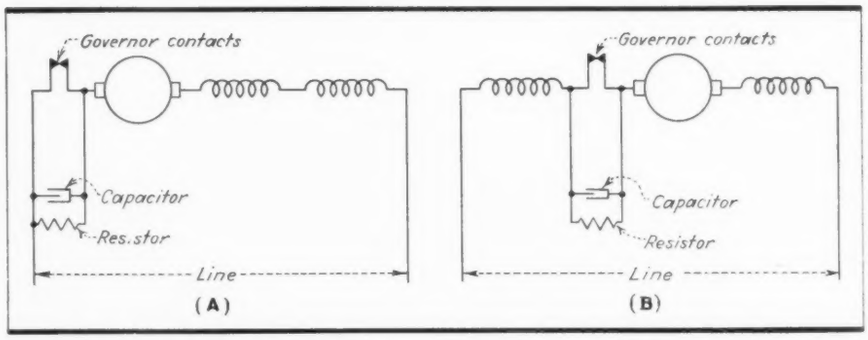


Fig. 8—Two typical connection diagrams for nonreversible governor-controlled universal motors. Increasing speed causes the contacts to open thus inserting resistance in the circuit causing the motor to slow down until the contacts close again, and the cycle repeats. The capacitor increases the life of the contacts. *(A)* connection for three wire motor; *(B)* for four wire motor.

D.C. Motors and Dynamotors. In addition to universal motors that can operate on d.c., there are four other major types of small d.c. motors: Shunt, series, compound, and split-series. Split-series is included as a basic type because of its present widespread use, particularly in aircraft work for control applications and for reversible actuators.

An interesting new application is an electrical torque meter currently being developed for aircraft engines that requires constant freqency power. To provide constant 400 cycle power, the small motor generator set of Fig. 9 uses a governor controlled constant speed d.c. motor driving a permanent-magnet alternator.

Another application of d.c. motors for instrument service is a small motor, only 3 1/4 inches in diameter, that accelerates a heavy flywheel from standstill, to 12,000 r.p.m. in 1/5 sec., developing 22 hp. at top speed. This is an exceptionally short duty cycle, and the output per pound of weight is unusually high.

Explosion-proof construction of small d.c. motors, such as used for a radar scanning application, is obtained by having a small volume of air inside the motor and by providing close clearances and long fits. If any explosion occurs inside the motor, the flame is sufficiently cooled by passing through the long thin passages that it cannot ignite any explosive mixture surrounding the motor.

Dynamotors, widely and extensively

used in radio work, are finding increasing usefulness. In effect, a dynamotor is to the d.c. system what a transformer is to the a.c. system; in short, it might be called a "d.c. transformer." A dynamotor is actually a d.c. dynamo and d.c. motor telescoped into one unit, hence the name dynamotor. Both the motor and generator windings are wound in the same slots, and have the same field structure. Power is fed into the motor winding through a commutator, usually at a low voltage; output is obtained from the high voltage winding by a second commutator, usually located at the opposite end.

In the conventional dynamotor just described, the output voltage at any load is closely proportional to the input voltage. To overcome this limitation, the special "Flying Tiger" type dynamotor shown in Fig. 10 supplies constant output voltage despite normal variations of the input voltage. This charactertistic is obtained by the use of two field structures and two armatures both on the same shaft. One armature carries both input and output windings; the other armature has only output windings. This second armature is, in effect, a booster armature. Regulating the field of the booster controls the output voltage. An automatic voltage regulator holds the output voltage constant by regulating the field of the booster.

In addition, this special dynamotor has three output windings, and hence, four commutators. Thus it has three output voltages, all of which are regulated so as to be essentially constant and independent of the voltage input over the usual range of input voltages. Only a single voltage regulator is used, and all three output voltages are controlled together. The dual armature from one of these unusual dynamotors with four commutators is shown in Fig. 3 while being balanced in the dynetric balancing machine.

THERMAL OVERLOAD PROTECTION. Overload protection is desirable in many small motor applications and is often an integral part of the motor. For example a typical thermally protected capacitor-start motor uses a built-in disc-type thermostat. The thermostat is located near the windings and responds both to heat received from the windings and to line current that flows through a heater in the thermostat. The thermostat heater also provides locked rotor protection. The motor can be locked on the line or operated at any degree of overload without danger of burning out the windings. When a dangerous temperature is approached, the thermostat disconnects the motor from the line. After the motor cools down to a safe temperature, the thermostat automatically re-starts the motor. When the thermostat is used to protect dual voltage motors, the heater is connected in series with only one of the two parallel circuits, but the contacts are connected in series with the line.

Thermal protection is also used on many small aircraft motors. The thermostat limits the maximum armature winding temperature, which in these motors is the limiting temperature, to approximately the same value at all ambient temperatures. Therefore the armature winding never exceeds a predetermined total temperature. In contrast, external current-operated devices allow the same temperature rise regardless of the ambient temperature.

As the nation swings into full peacetime production an increasing number of new and redesigned products will require small motors. The choice of the right fractional horsepower motor will be merely a matter of matching the performance characteristics of the motor to the starting and running needs of the product. This will be relatively easy since standard motors offer such a wide range of starting and running characteristics. Although peacetime products will vary widely, designers will generally find suitable motors for their new and redesigned products among the diversified types of standard fractional horsepower motors.

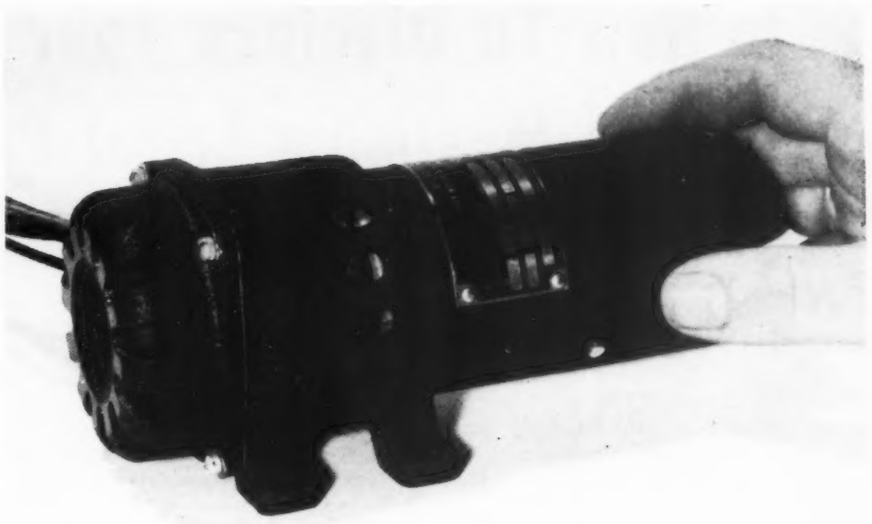


Fig. 9—Constant-frequency motor generator set for supplying power to a torque meter for airplane engines. A governor controlled d.c. motor drives a permanent-magnet alternator to produce an unvarying 400 cycle power supply.

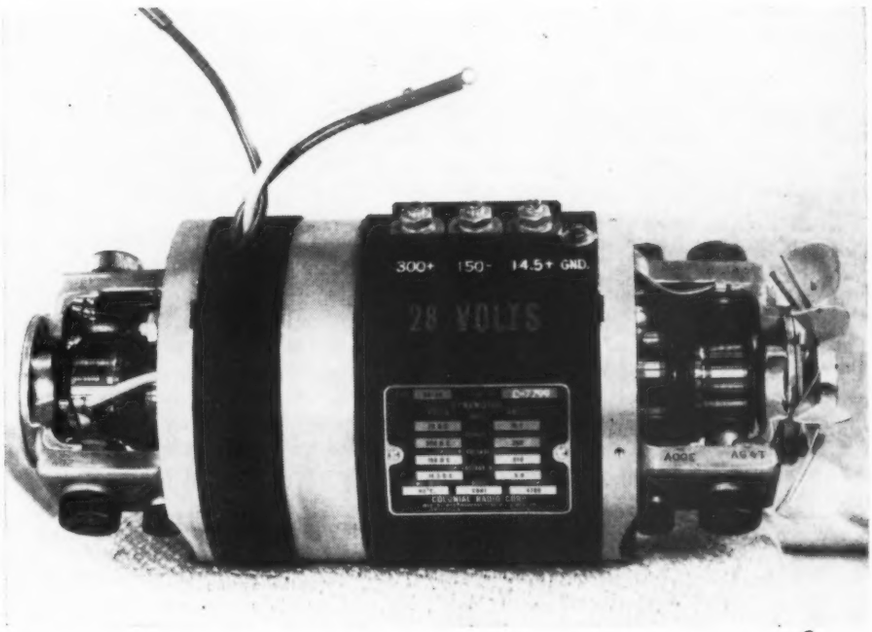


Fig. 10—By regulating the field of a booster built into this dynamotor, three output voltages are maintained practically constant during ordinary changes of the input voltage. Usually changes in input voltage are reflected in output.

How To Disclose Your Invention For Maximum Legal Protection

WILLIAM K. RIEBER
Patent Engineer

RECENT FIGURES from the Official Gazette of the United States Patent Office indicate clearly that the number of issued patents is on the upswing. Unfortunately, however, the majority of infringement cases are being decided against the patentee. This would seem to indicate that the increase in quantity of patents is accompanied by a decrease in quality. Various reasons have been assigned for this trend; among them, a change in attitude on the part of the Patent Office and on the part of the courts. It is quite evident, however, from an examination of recent cases, that many of the poor patents issued recently are the result of inadequate liaison between the inventor and his attorney.

The largest part of the expense of obtaining a patent is usually the attorney's fee, based on the amount of time he spends on the case. Thus, it is common sense for the inventor to make the most of his attorney's time. To avoid wasting time in needless questioning and research on the part of the attorney, the inventor should know the subject thoroughly. Generally speaking, the technical aspects of an invention are the problems of the inventor, while the legal aspects, matters of form, and the conduct of business with the Patent Office are the responsibility of the attorney.

Unfortunately the enthusiasm for a new idea sometimes blinds the originator to possible shortcomings. Analysis of one's own invention is in some ways analagous to a parent's criticism of his own child. The inventor should therefore attempt to look at his "brain child" from an unbiased and detached point of view, recognizing faults as well as advantages.

In first presenting an invention to an attorney, the record of invention, including the evidence of conception, notes as to reduction to practice, laboratory notes and a model, are of considerable assistance, but they do not present the entire picture. Depending on a model as the only means of disclosing invention to an attorney is frequently a mistake. While models are of some assistance in illustrating the physical principles involved in an invention, they are usually not representative of the finished product in its best form. Furthermore, models are frequently bulky and awkward to handle, and few attorneys have adequate facilities for storing them in their offices. It is usually preferable, after the attorney's first examination of the model, to substitute a photograph, properly annotated, which can be filed or sent through the mail.

The best form of presentation is a complete description accompanied by drawings or sketches. Such descriptions are referred to by various titles such as "Amateur Specifications", "Inventor's Disclosure" and "Preliminary Disclosure". It will be referred to hereafter as a Preliminary Disclosure. A typical Preliminary Disclosure and its accompanying drawings are illustrated on these pages. This form is illustrative only, since no one form could cover all types of invention.

The drawing can be a pencil sketch but should be carefully executed and large enough to show all the parts clearly. It is wise, while making the drawing as large as possible, to keep it letter size (8½ in. x 11 in.). It is also advisable to make the drawing on tracing paper in order that several copies can be made.

Working drawings are frequently more of a hindrance than help in illustrating an invention. The notes to machinists, tolerances, detailed data as to heat-treatment, plating or other shop processes, and the meticulous dimensioning of working drawings only clutters them up from the attorney's point of view. Photographs alone are also inadequate illustrations for the preliminary specification. It is difficult to make notes and reference figures on them and frequently they fail to show clearly the most important elements of the invention.

Since the attorney is accustomed to reading and describing drawings made according to patent office requirements, it is usually advisable to follow this practice to some extent in making the sketches to accompany a preliminary disclosure. One practice observed in Patent Office drawing that it is wise to follow, is the use of numbers rather than letters in designating the various parts of the drawing. It is customary to number the parts in the order that they appear in the description, starting with a number larger than the number of figures. If this procedure is observed, section lines can be designated with the number of the figure showing that particular section. It is also advisable to avoid the use of dotted lines to show hidden surfaces except where this is essential to illustrate some point in the description. A brief outline of the conventions used in patent office drawing is given in the Rules of Practice which may be obtained, free, from the Patent Office.

The Preliminary Disclosure should, in general, follow the conventional form used in the preparation of patent specifications. This form usually should be made out as follows:

1. Delineation of the art in which the invention falls, giving the general and particular fields of use.
2. A discussion of the prior art involved, with particular reference to disadvantages, inadequacies and problems that resulted in the present invention.
3. The objects of the invention specifically enumerated. It is wise to avoid such vague generalities as: "It is an object of this invention to provide an improved mouse trap". Rather, one should say: "It is an object of this invention to provide a mouse trap which does not require resetting. It is another object of this invention to provide a mouse trap in which the bait cannot be stolen."
4. A brief description of the various figures of the drawing.
5. A detailed description of the construction shown in the drawing. Alternate forms and alternate materials may be just as important or even more important than the forms and materials specifically illustrated. For this purpose, it is wise to use such an expression as this: "While it was necessary to make gasket 7 of non-inflammable material in the machine illustrated, it might and probably does have many other applications in which it would not need to be non-inflammable. In these other applications, a gasket constructed of cork or fibre, paper, etc. could be used."
6. A closing paragraph or two showing how the invention accomplishes the objects originally set forth. Again it is well to remember that there may be many other uses and forms which the invention may take and to try to anticipate all of these.

Self-Appraisal

It would seem obvious that the inventor should be thoroughly familiar with the art in which his invention lies. This, unfortunately, is frequently not true. Many an inventor adopts an ostrich-like attitude and is afraid to look at the related art for fear that his alleged invention may be found to be anticipated. If, at the suggestion of the attorney, a search is made before preparing

HYDRAULIC TACHOMETER

Preliminary Disclosure

This invention relates generally to the measurement of rotary speeds, and more particularly to sensitive tachometers which take no power from the apparatus under test.

Tachometers usually make use of either centrifugal force operating through a mechanical linkage, or magnetic drag working against a spring. The first method is inaccurate in that friction losses are introduced, and the second method involves the use of a permanent magnet the strength of which may change rendering the measurements inaccurate.

One object of the present invention is to provide a tachometer which contains no mechanical parts.

Another object is to provide a tachometer in which the characteristic deflection does not change with age.

Still another object is to provide a tachometer unaffected by surrounding magnetic fields.

In the attached drawings,
Fig. 1 is an isometric view of a tachometer constructed according to the invention and mounted on an electric motor under test,
Fig. 2 is a section taken on the line 2-2 in Fig. 1,
Fig. 3 is a section through one modified form of expansion chamber used in the invention, and
Fig. 4 is a second modified expansion chamber.

In the present embodiment, the device has a T-shaped body 5 with an opening in the lower end which can be clamped like a collar onto the rotary shaft whose speed it is desired to measure.

The sensitive elements are two expansion chambers 10-10, which are interconnected by a passage through the body 5. The chambers 10-10 are filled with a liquid 8 and may be sylphon bellows as shown in Fig. 2, aneroid cells 14 as shown in Fig. 3, or may be chambers in the body 5 covered with thin diaphragms 15 as shown in Fig. 4. In any case, the chambers are so arranged that the centrifugal forces occasioned by rotation of the body 5 about its longitudinal axis cause expansion of the chambers. If desired, weights 7-7 may be attached to the chambers to increase the centrifugal forces. In the form shown in Fig. 4, no weights are used, the weight of the liquid being relied on to expand the chambers by bulging out the diaphragms 15-15.

In order to accurately measure the change in volume of the chambers, a capillary tube 9 is employed. The tube 9 is mounted co-axially with the body 5 and connected to the passage which joins the chambers. A sufficient quantity of the liquid 8 is placed in the chambers to bring the level up into the tube where it may be read against a stationary scale 12 as indicated by the numeral 11 in Fig. 2. The position of the liquid level 11 will depend on the degree of expansion of the chambers 10-10, which in turn is dependent on the speed of rotation of the body 5.

GENERAL OBSERVATIONS

The use of relatively heavy liquids ilke mercury and carbon tetrachloride increase the sensitivity at low speeds.

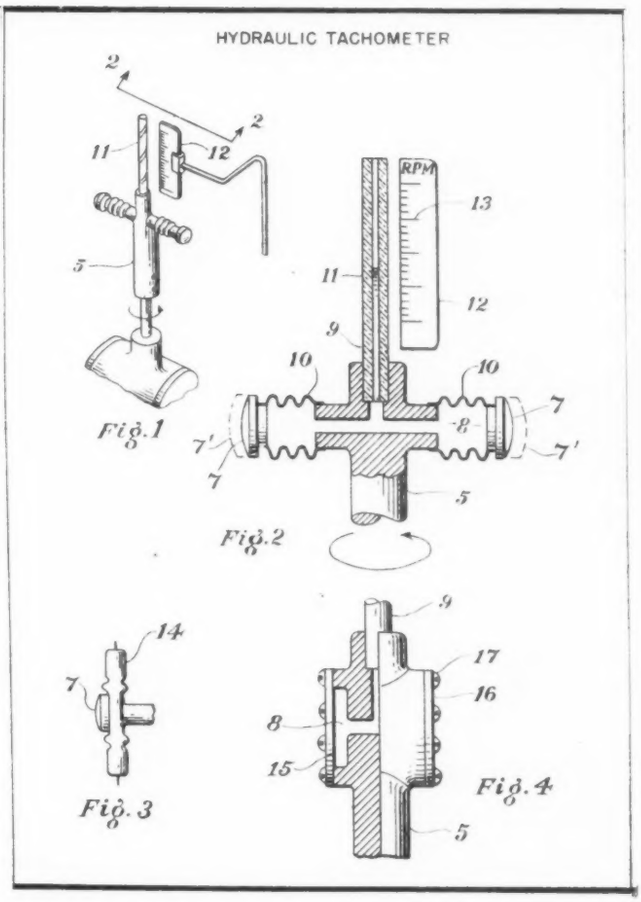
The expansion chamber should preferably be made of relatively iso-thermal material — although steel, brass, phosphor bronze, beryllium copper, and similar materials may be used.

The liquid 8 should be colored for ease of observation.

The scale divisions 13 may be engraved or etched on the tube 9.

Provision should be made to slide the scale 12 up and down to "zero" it before taking readings.

Motion of the liquid in the tube 9 may be used to actuate governor controls.



Sketches should be made in accordance with regular patent office procedure, as shown above. Reference numbers should be fully explained in the descriptive section Preliminary Disclosure, shown at the left.

and filing an application, the inventor should read the "references" (prior patents in the related art) and understand them thoroughly. In reading prior patents found by the searcher, particular attention should be paid to the drawings and the description, the claims being of no particular value in a "pre-ex" (preliminary examination) search. It should be remembered that the purpose of the pre-ex search is to discover anticipation rather than to decide whether any patents may or may not be infringed.

The alleged invention illustrated in this article was well and truly anticipated; that is, somebody else thought of it first. Two of the most pertinent references cited against the "Hydraulic Tachometer" are reprinted in part in Figs. 2 and 3. The reader may wish to test his ability to read patents by examining the cited patents to find the various elements analogous to those in Fig. 1.

As a final aid to the attorney, the inventor should carefully check over the specification and claims before the application is filed. Again, he should keep in mind that matters of form and phraseology, are definitely problems of the attorney, the inventor should devote his efforts to a careful analysis of the claims.

During the prosecution of the application, the inventor should study all actions which relate to the merits of the invention. If, for example, a rejection by the Examiner is based on some prior art, the inventor should examine each reference carefully and attempt to aid the attorney in pointing out why this reference is not pertinent if such be the case.

Often the prolific inventor will have several patent applications in process simultaneously. Usually these will relate to the same art, possibly one of the applications being a basic or generic invention and the others being in the nature of improvements. In order to avoid the harmful effects of a practice known as "double patenting," the inventor should check each application in its relation to the others before such application is allowed to issue, and if possible, the various applications should be so maneuvered that the basic or generic patent issues first. A careful attorney will always watch this, but it occasionally happens that an inventor may employ different attorneys for different applications, and thus a patent may issue without the attorney realizing that there are other applications to consider.

During the preparation, filing and prosecution of the patent application, the inventor should of course make it perfectly clear that his services are intended to augment and not replace the functions of the attorney. The competent attorney will recognize the value of the inventor's assistance given in this way, and a much more valuable patent will result.

NEW MATERIALS AND PARTS

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Tube, Thyratron	Sylvania Electric Products, Inc.	434
Valve, Solenoid	Allied Control Valve Co.	434
Wheel, Blower	The Lau Blower Co.	433
Wheel, Industrial	Thermoid-Grizzly Wheel Sales	432
X-ray Diffraction Tube	General Electric X-ray Corp.	433
X-ray Scanner	North American Phillips Co., Inc.	431

Viscous Torsional Vibration Damper

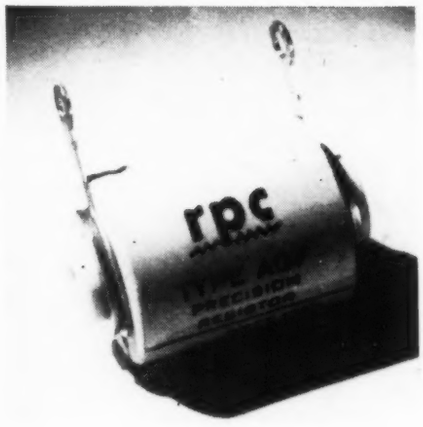
Houdaille-Hershey Corp., Buffalo 11, N. Y.

This unit consists of three parts: the damper mass, its housing and a bronze bushing. The viscous substance is Silicone fluid. The housing is hermetically sealed to retain the fluid. This device is a damper and not a detuner. The housing is attached to the engine crankshaft and the inner, free-running damper mass is separated from it by a thin film of the Silicone fluid. Because of the high viscosity of this fluid, the damper mass rotates at engine speed. When the plus and minus changes in velocity which produce vibration are transmitted from the crankshaft to the housing, the damper mass, being heavy in comparison, tends to continue to rotate at constant speed. The vibrations are damped by the drag of the viscous fluid separating the two parts and its resistance to the shearing action.

Wire Wound Precision Resistors

Resistance Products Co., 140 S. Second St., Harrisburg, Pa.

These resistors are protected against corrosion by a double impregnation of special insulating varnish. Labels of cellulose acetate protect the windings. The resistors can be mounted by the terminals or by means of a 6-32 machine screw through the center hole of the unit. Type AFB is 9/16 in. diameter by 15/32 in. long. Type AGC is $\frac{5}{8}$ in. diameter by $\frac{5}{8}$ in. long. Type AGF is $\frac{5}{8}$ in. diameter by 1 in. long. Type AJS



is $\frac{3}{4}$ in. diameter by 1 9/32 in. long. Type ALP is $\frac{7}{8}$ in. diameter by 2 1/16 in. long. Maximum resistance with use of .0014 in. diameter nickel-chrome alloy wire is as follows:— Type AFB, .225 meg., Type AGC, .450 meg., Type AGF, .800 meg., Type AJS, 1.50 meg., Type ALP, 2.75 meg. Standard resistance tolerance is 1 percent, but tolerances of $\frac{1}{2}$, $\frac{1}{4}$ or 1/10 percent can be furnished.

Micro Conductivity Cell

Industrial Instruments, Inc., 17 Pollock Ave., Jersey City 5, N. J.

This dip cell is made of Pyrex glass, and measures $\frac{1}{2}$ in. outside diameter by 6 in. long. The platinized platinum electrodes are so positioned that only a few cubic centimeters of sample are required. A flexible rubber sleeve protects the lead wires. A selection of cell constants from 0.1 permits the checking of any solutions ranging from pure distilled water to fuming sulphuric acid, when the cell is used in combination with a suitably calibrated solu-bridge or a conductivity bridge.

Bellows-Type Pressure Gage

Manning, Maxwell & Moore, Bridgeport, Conn.

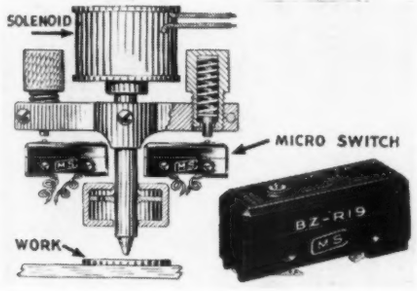
This low range pressure gauge is suitable for indicating draft pressures or any low pressures of gases or liquids that are not corrosive to bronze. The pressure element is a self draining bronze bellows with a phosphor bronze calibration spring. Adjustable stops protect the bellows from excess pres-

sure or vacuum. It is supplied in 4½ in. and 6 in. phenol cases with ¼ in. male pipe thread lower or back connection. Type 1188 P is supplied in pressure ranges from 10 in. of water to 10 lbs. pressure. It is also supplied as a low range vacuum gauge Type 1188 V in vacuum ranges from 10 in. of water to 20 in. of mercury vacuum.

Part Sorting Switch

Micro Switch Div., First Industrial Corp., Freeport, Ill.

A switch for the automatic sorting of small parts. When the solenoid is de-energized the plunger assembly descends by gravity under control of the air dash pot. The



point at which the plunger is stopped by the work beneath determines whether neither switch or one or both switches are operated. A timer in the electrical circuit controlled by these switches operates an air blast to blow the work into "OK" or "rejection" bins. Electric lamps may also be used for visual indication of results. This switch is claimed to have a repeatable accuracy of 0.0001 in.

Sealing Compound

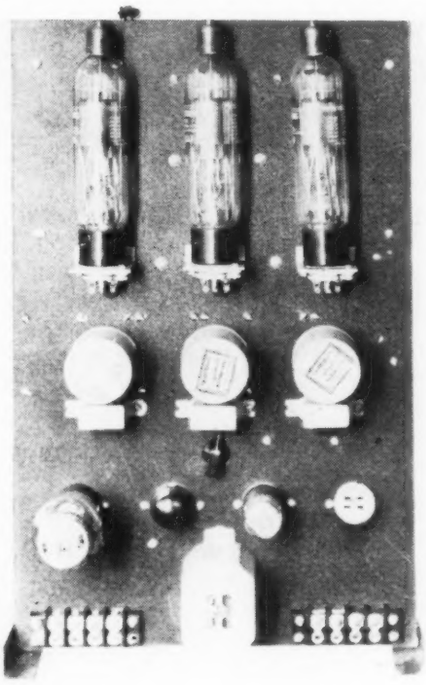
Parker Appliance Co., Cleveland, Ohio

A general-purpose thread and gasket sealing compound, known as Uniseal. This organic product produces a seal against air, water, steam, gas, gasoline, oil, hydraulic fluids, and the aromatics. Testing is claimed to have shown virtual insolubility in gasoline, kerosene, Stoddard solvent, benzene, xylene, carbon disulfide, the ketones, acetones and the various aromatic aviation fuels. The product is a paste containing no free metallic particles. The compound flows to form ribbon gaskets, and blends with cut gasket material. Uniseal is effective under conditions of vibration, and at temperatures above 300 deg. F.

Variable Speed Drive

Submarine Signal Co., 160 State St., Boston 9, Mass.

This unit, Servotron, operates an alternating current and through an electronic controller runs a direct current motor at speeds which are infinitely variable within the

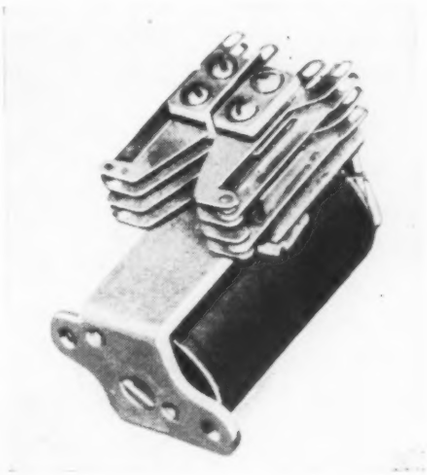


motor's speed and load capacity. It has dynamic braking and reversal without speed overrun. A line of drives from 1/50 h.p. to 5 h.p. is available.

Multi-Pole Type Relay

R-B-M Division, Essex Wire Corp., Logansport, Ind.

A small a.c. multi-pole relay with shunt coil for electronic and communication applications. Standard voltages range from 1½ to 220 volts 60 cycle. Contacts are available in from one to four poles, normally open, normally closed or double break. The double throw contacts can be "break-before make" or "make-before break". Maximum contact rating—non-inductive, 3 amperes—24 volts a.c. 1 ampere—110 volts a.c. Coil rating is 11 volt-amperes—inrush; five volt-amperes sealed. This relay is also available in single pole normally open double break contact, 10 ampere rating, at 115 volts. a.c. The approximate overall dimensions are: 1-61/64 in. long; 1-34/64 in. high (four

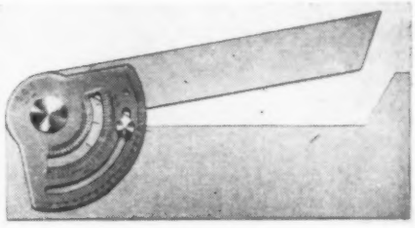


pole double-throw); 1-35/64 wide. The heel piece of the magnet frame has two tapped holes for rear mounting and two pierced holes for front mounting. Solder type terminals.

Drafting Instrument

Stewart-Jackson Instrument Co., A. G. Bartlett Building, Los Angeles 14, Calif.

An adjustable quadrangle combining drafting features not usually available in one instrument. Angles from 0 deg. to 90 deg.; pitch scales from 0 to 24/12; percentage

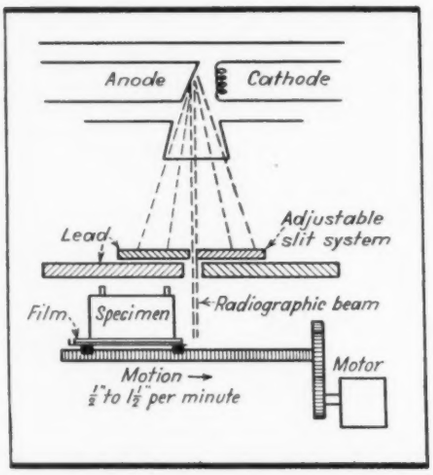


slopes from 0 percent to 100 percent; sine or cosine functions and tangents may be found with the S&J Quadrangle. This instrument has eight drawing edges, is rectangular in shape, and may be used as a triangle. The instrument measures 4 in. x 11 in.

X-ray Scanner

North American Philips Co., Inc. 100 E. 42nd St., New York 17, N. Y.

The scanner unit measures 32 in. long, 14 in. wide, and 16 in. high and consists of a motor-driven table on which the specimen and film move beneath an adjustable slit through which the X-rays pass. Table speed is ½ to 1½ in. per minute and can be



adjusted to suit each particular application. The scanner records the geometric relationship of integrated components and spacings of segmented specimens. When fine grain X-ray film is used, enlargements may be made for more accurate determination of element position and condition.

Calculating Device

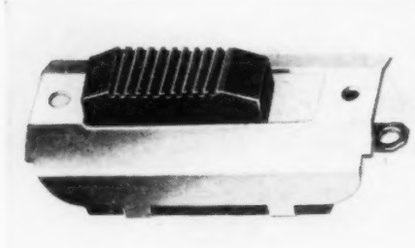
Heat-O-Meter, 424 West 42nd St., New York 18, N. Y.

It is claimed this Heat-O-meter, a round dial with three concentric celluloid printed disks, eliminates the figuring usually necessary to determine radiation. This dial contains sizes of mains, returns, risers, radiator sizes and capacities, round and sectional boiler net ratings, chimney flue sizes and capacities with minimum and maximum heights, hot water tank sizes and capacities, fuel oil tank sizes and capacities, hot water generator capacities and other heating information.

Torsional Spring

Allied Control Co., Inc., 2 East End Ave., New York 21, N. Y.

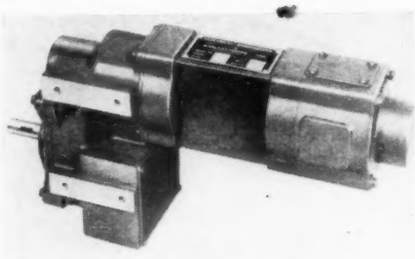
A torsional spring arrangement designed to assure snap. The switch has a formed steel housing and heavy laminated Bakelite base. It is actuated by a slide-type molded Bakelite button. The E-4 switch is supplied as single-pole double-throw, single-pole single-throw normally open, or single-pole single-throw normally closed. It is rated at 300 watts a.c., 3 amperes at 115 volts a.c., ¾ amperes at 115 volts d.c. non-inductive. The switch is 1 5/6 in. long, ⅝ in. high, 9/16 in. wide and weighs approximately ½ oz.



Automatic Door Actuator

Electrical Engineering and Mfg. Corp., 4606 W. Jefferson Blvd., Los Angeles, Calif.

This unit is designed for instant reversing and can be overridden in case of power stoppage. It is built for resistance to damage from vibration and has built-in thermal protection against damage from overload when the door to which it is fitted is accidentally blocked. Starting torque is 250 in. lb.; average torque at 105 r.p.m. is 170 in. lb. Unit operates on 32, 64, or 110 volts d.c. and can be modified to operate on



110-volt, single-phase a.c. It has a magnetic clutch, and three limit switches operated by cams that can be adjusted to modify the characteristics of the unit to meet various conditions.

Layout Protractor

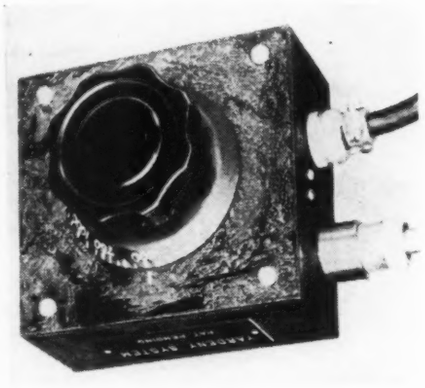
Engineers Specialties Div., The Universal Engraving and Colorplate Co., 980 Ellicott St., Buffalo 8, N. Y.

Designed for extremely accurate layout work. The ruling and graduations are on the underside of the quarter-inch beveled plate glass of which the instrument is made. The circular edge of the glass is beveled to a thin outer edge through which the divisions are viewed. The graduations designate ten minutes of arc on the outer scale. This protractor has cushioned metal handles and measures 13¾ in. x 7½ in.

Remote Control

Yardeny Laboratories, Inc., 105 Chambers St., New York 7, N. Y.

The dial can be set to any desired speed and the speed changer will follow as fast as the electric servo motor can drive the speed adjusting screw. The dial can be calibrated in r.p.m., ft. per min., or gal. per hr. It is claimed that any speed within the range of the changer can be selected with an accuracy of 2 percent. In operation, Flectrol controls the rotation of the small

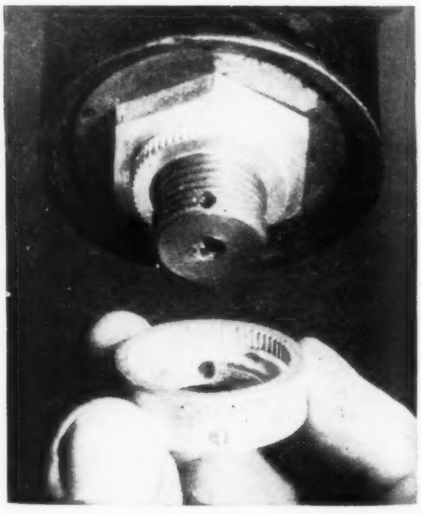


motor used to adjust the pulleys or gears of the speed changer. Rotation of the speed selector dial causes this motor to drive the adjusting screw until the output speed of the controlled device corresponds to the speed selected on the dial.

Adjustment Nut

E. S. Firestone Engineering Co., 11127 Magnolia Blvd., North Hollywood, Calif.

The Pin-Point-Adjustment Nut permits micrometer settings to 0.001 take-up on 16 pitch threads with considerably finer adjustment possible on threads with increased

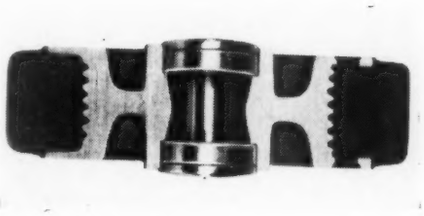


pitch. It is adaptable to securing proper spindle bearing adjustment to eliminate wheel wobble and achieve better wheel alignment.

Industrial Wheel

Thermoid-Grizzly Wheel Sales, Chicago, Ill.

These wheels have automatic lubrication to the sealed bearings. Holes in the casting allow the rubber tire to expand under pressure, and contract when the pressure is removed. It is claimed that the cut-resistant

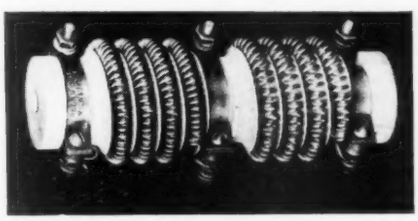


rubber tread will not pull apart under any conditions and will not cut or mar floor surfaces. The "Grizzly Wheel" is available in standard diameters of 6, 8, 10, and 12 in.

Center-tap Resistor

Techtmann Industries, Inc., 828 N. Broadway, *Milwaukee* 2, *Wis.*

The resistor consists of a core, machined from a solid piece of steatite, upon which is wound a helical resistance wire element. Center and end taps are half-straps clamped to the core. The tap assembly has brass



nuts and bolts and stainless steel lock washers. All parts are zinc plated. Protection against fungus growths is provided by an additional chromate dip plating.

Flash Capacitor

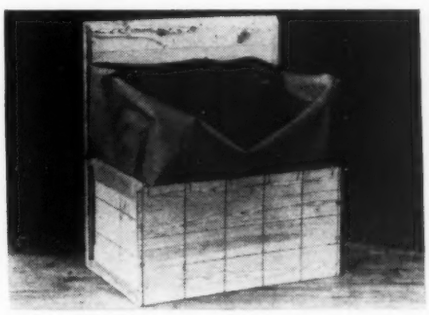
Tobe Deutschmann Corporation, Canton, Mass.

With energy storage capacity of 100 watt seconds at a peak rating of 2500 volts d.c., the "N-Erg-Y" capacitor is designed for use with portable speed-flash units. It is contained in a hermetically sealed steel case measuring 6½ in. x 4⅝ in. x 3¾ in. and weighs 6¼ lb. An expected life of 10,000 charge-discharge cycles at peak rating, as well as ability to hold the peak charge for approximately twenty-four hours is claimed for this unit.

Packing Liner Fabric

Protective Coatings Corp., 689 *Main St., Belleville* 9, *N. J.*

Aquastop will withstand exposure to changes in temperature from 20 deg. F. to 175 deg. F. at maximum relative humidity, without cracking, tackiness, flow, loss of flexibility or waterproofness. All seams are caulked with Aquastop cement which effects a "chemical weld" of adjacent Aquastop liner panels thereby producing a sealed waterproof barrier between the contents and the exterior wooden shipping box. The cemented

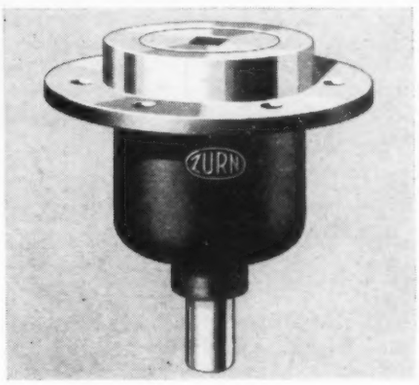


seams are as strong and as waterproof as the liner material. When required, Aquastop can be heat sealed at temperatures between 265 deg. F. and 310 deg. F.

Remote Valve Control Assembly

J. A. Zurn Mfg. Co., Erie, Pa.

These assemblies actuate valves located below floor level, close to the ceiling or walls and partitions, and inside tanks or processing vessels. The essembly consists of a flexible cable, for transmitting torque to the handle of the valve to be operated, or a rod or pipe connected by universal joints; a remote valve control box; sleeve bushings, terminal brackets, cable clips and a rod coupling for the valve wheel. The valve control boxes are made in cast bronze, steel, semi-steel or iron of various dimensions, and can be attached to masonry construction, metal or wood partitions.



Flexible Diaphragm Cloth

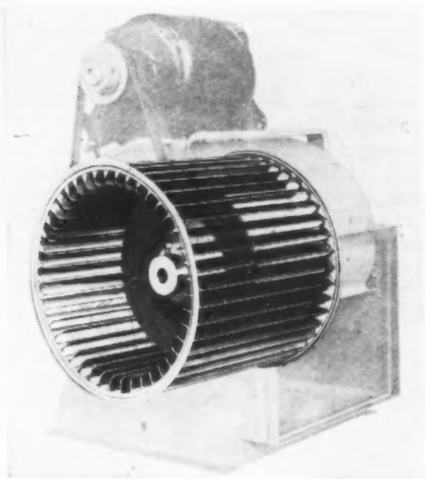
Irvington Varnish & Insulator Co., 6 Argyle Terrace, Irvington 11, *N. J.*

The 80/80 base, Style DL-011 diaphragm cloth has good mechanical strength, and it is claimed that this cloth retains its flexibility and is practically unaffected by wide temperature changes. It is available in standard 100- to 500-yd. rolls, approximately 37 in. wide.

Blower Wheel

The Lau Blower Co., Dayton, Ohio

The Lau Series "A" Blower Wheel is a center suspension wheel. It is claimed to have greater mechanical strength, truer concentricity and balance, and more efficient



performance than ordinary types of blower wheels. Each blade is mechanically locked to the center disc. The end ring is narrower to permit greater air flow into the wheel.

X-ray Diffraction Tube

General Electric X-Ray Corp., 175 W. Jackson Blvd., Chicago 4, *Ill.*

These diffraction tubes have a target construction which increases the heat loading by presenting the grain structure of the anode material in a dense condition at right angles to the surface. This is claimed to extend the useful life of the tube. Diffraction tubes are available with 7 types of targets: Tungsten, molybdenum, copper, nickel, cobalt, iron and chromium. Silver and manganese, can be supplied on special

order. The increasing demand in diffraction work for use of longer wavelengths has brought about this multiplicity of target requirements. Development is proceeding on titanium and germanium. Various alloy targets have been built for special purposes.

Solenoid Valve

Allied Control Valve Co., Inc., 50 Norwalk, Conn.

This unit weighs 1 lb., has a power consumption of 10 watts and operates under pressures to 150 lb. per sq. in. It is a three-way, two-position valve designed for use with water, alcohol, hydraulic fluids, or air and is made for a.c. or d.c. operation and for intermittent or continuous duty. All parts in contact with the fluid handled are of stainless steel; the coil is impregnated to withstand moisture and the stems are welded.

Thyratron Tube

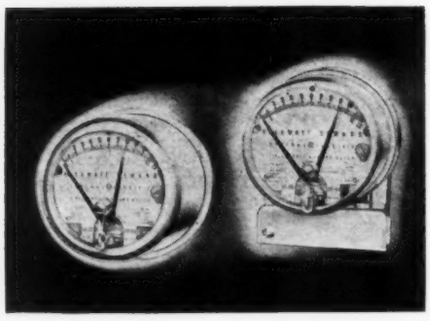
Sylvania Electric Products Inc., Radio Tube Div., 500 Fifth Ave., New York 18, N. Y.

This thyratron, measuring 2⅛ in. overall and mounted in a T-5½ bulb is designed for electronic control applications where equipment must be compact, portable or lightweight. The tube may be operated in any position and is not affected by ambient temperature. Typical ratings and characteristics of the type 6D4 miniature thyratron are: heater voltage, a.c. or d.c. 6.3 volts; heater current 250 milliamperes; maximum voltage between elements 450 volts; heating time 30 seconds; maximum peak anode current 100 milliamperes; maximum heater-cathode voltage — 100 volts + 25 volts.

Thermal Watt-demand Meter

General Electric Co., Schenectady 5, New York

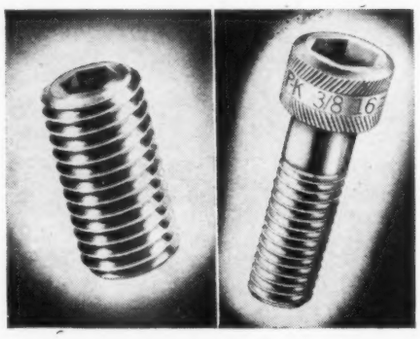
This meter, operates on the "direct-heat" principal and is known as Type HI-1. The "direct-heat" principle is based on the use of temperature-sensitive, bimetallic spirals that act as their own heaters. The temperature-sensitive elements are attached directly to a shaft and gear. As the electric energy flows through the thermal element, the heat produced causes the spiral to deflect, thus rotating the shaft and gear, which drives a red pusher pointer. Maximum demand is indicated by a black pointer, which is pushed up-scale by the red pointer. The meter is housed in a glass case, coated on the inside except for the circular front opening, to promote uniform heat distribution within the meter and to aid in reducing effects of variable external temperatures. It has a manually-operated reset device, by which the black pointer can be turned down to contact the red pointer. The scale conforms to the standards approved by the joint committee of E. E. I. and A. E. I. C.



Socket Screws

Parker-Kalon Corp., 200 Varick St., New York 14, N. Y.

The threads are ground on hardened stock, with no subsequent heat-treatment required. The socket head cap screws have the size marked on the head. The "gear grip" head design is claimed to prevent slipping and save time.



Lightweight Plastic

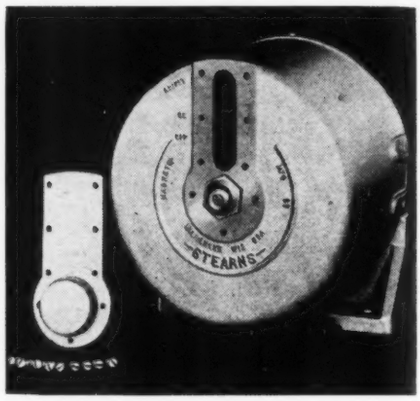
E. I. duPont de Nemours and Co., Inc., Wilmington, 98, Del.

Called cellular cellulose acetate, or "CCA," this plastic combines insulation against heat and cold with structural strength when bonded between two sheets of metal, wood, or plastic. The thermal insulating properties of cellular cellulose acetate are virtually the same as those of cork, balsa wood, and other rigid insulating materials. The heat resistance of cellular cellulose acetate is such that thermosetting resins can be used to cement it. The material is uniform in density. It is not brittle, nor will it crack or break down under vibration. The plastic is made in strip 3½ in. wide, ⅜ in. thick, and of any desired length.

Magnetic Disk Brake

Stearns Magnetic Mfg. Co., Milwaukee 4, Wis.

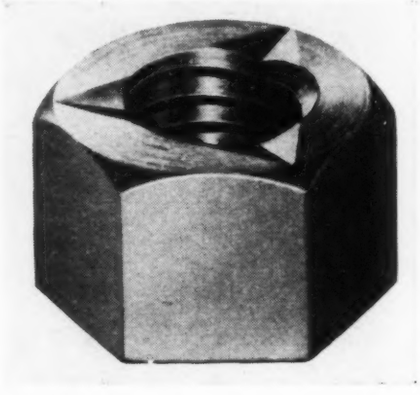
The Series 1300 magnetic disc brake, designed for use on applications to 100 h.p. motor driven loads can be supplied in either floor or motor mounted types and for either alternating or direct current. The design permits replacement of any part without disturbing the brake assembly.



Lock Nut

Grip Nut Company, 308-J S. Michigan Ave., Chicago 4, Ill.

The Gripco lock nut is a semi-finished, one-piece, self-locking nut of standard dimensions, not affected by oil, water or chemicals. It requires no lock washers or cotter pins. The triangular deflections on the top provide a controlled friction lock. The nut is free-spinning until the bolt threads contact the deflected threads near the top of the nut.



Silicone Rubber

Dow Corning Corp., Midland, Mich.

Silastic, the heat-stable silicone rubber, when properly cured, has physical and mechanical properties which recommend it for industrial applications in which thermal stability and oxidation resistance, combined with waterproofness and high surface resistivity are important factors. Silastic can be molded or extruded into any required shape. It can be applied as a coating for glass cloth, metal or wire. It is useful as an electrical insulating material and is a rubber-like bonding material for the fabrication of heat-stable, flexible and waterproof laminations of glass or asbestos cloth.

Charts for Fluid Friction Bearing Design—II

H. W. HAMM

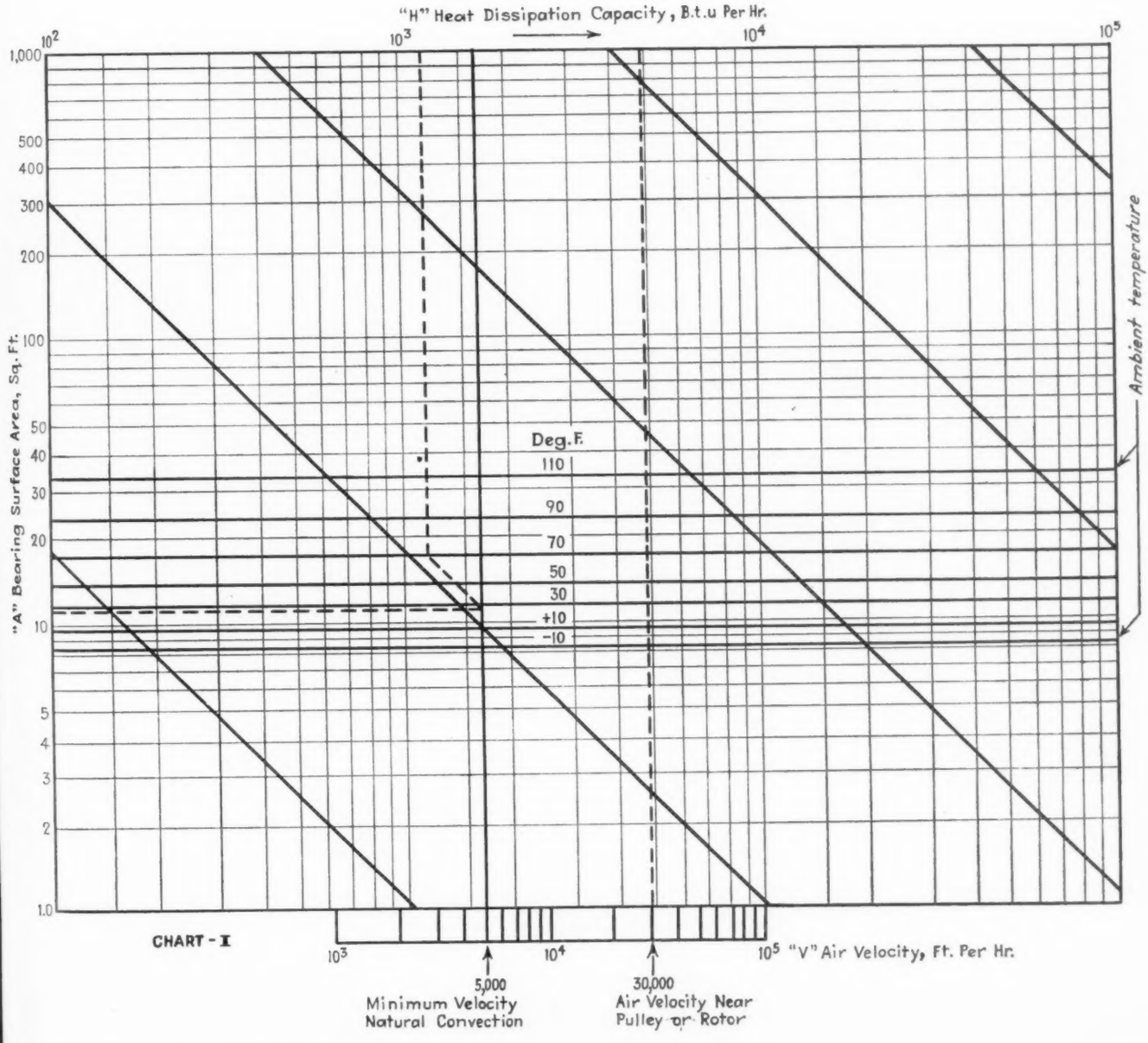
These charts, a continuation of a previous Reference Book Sheet, aid in the design of bearings operating under fluid friction conditions.

Chart I is for the air cooling of bearings and other oil-containing housings, and is a graphical presentation of the formula

$$H = 1.37 \times 0.015 \times V^{0.6} \times A^{0.8} \times \Delta T$$

where H = the heat dissipation capacity in B.t.u. per hr.
V = the air velocity in ft. per hr.
A = $K \times$ shaft dia. $\times$ sleeve length = bearing surface in sq. ft.
ΔT = the ambient temperature in deg. F.
K = 40 to 50

(Continued on next page)



Charts for Friction Bearing Design (continued)

Chart II is a graphical presentation of Petroff's Law for guide and load-carrying sleeve bearings operating under fluid friction. This relationship is expressed by the formula

$$H = \frac{D^2 \times D/C \times L \times N^2 \times Z}{1.04 \times 10^9}$$

where H = the friction loss in B.t.u. per hr.
D = the shaft diameter in inches
C = Diametral clearance in inches
D/C = the clearance-diameter ratio
L = the bearing sleeve length in inches
N = the r.p.m.
Z = the oil viscosity in centipoises

It is to be noted that this law is valid when the bearing constant $S_z \leqq \frac{Z \times N}{p} = 100$

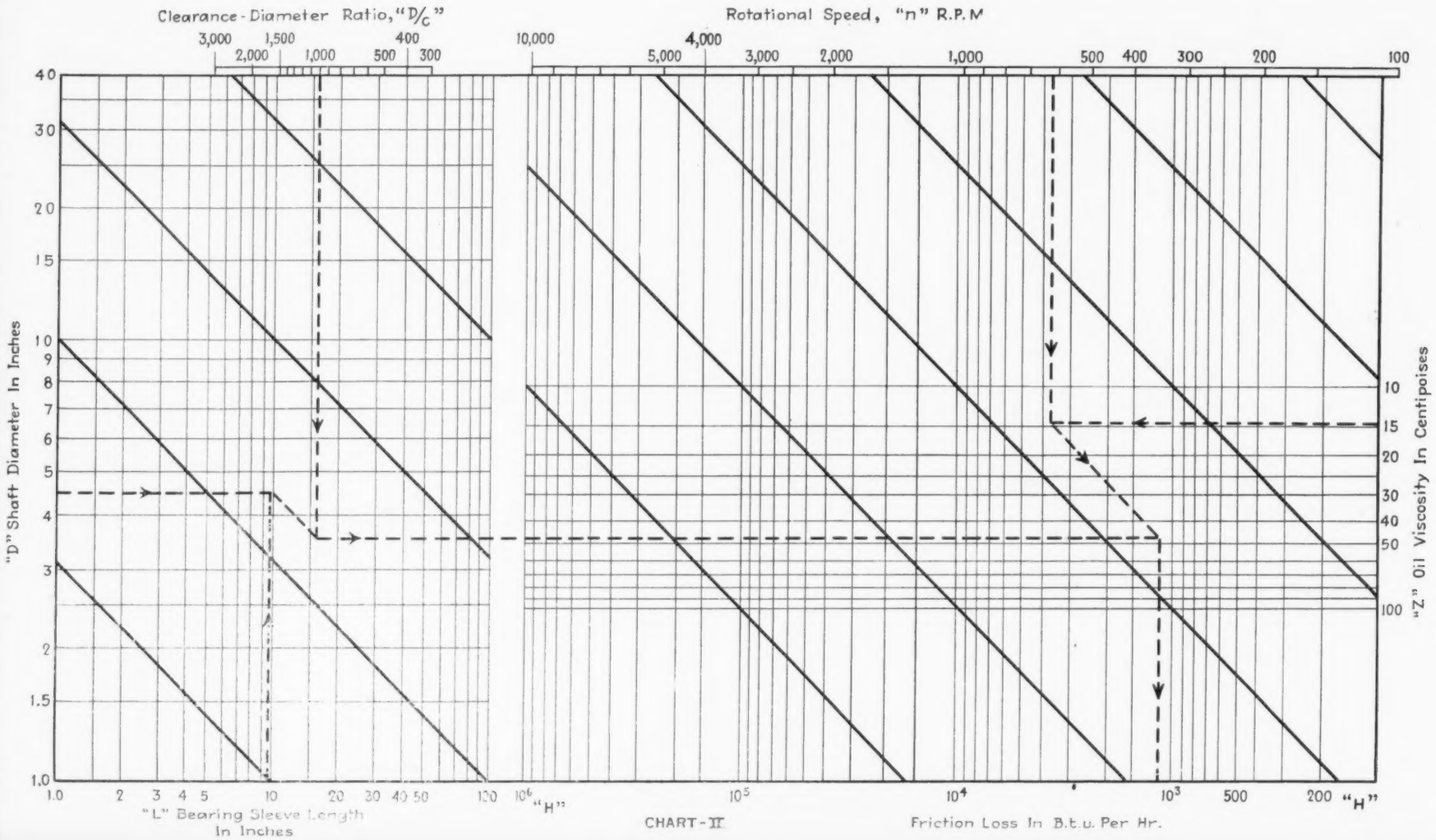
where p = the unit load in lb. per sq. in. on the projected area.

Additional information on the design of bearings for fluid friction operation may be obtained from the following sources:

"Computations for Cooling by Oil-Hydraulic Means," PRODUCT ENGINEERING, March, 1944, page 151.

"Vertical Guide Bearings," *Electrical World*, March 3, 1945, page 119; March 17, 1945, page 108; March 31, 1945, page 103.

"Hydro-dynamic Formulas Useful in Bearing Design," PRODUCT ENGINEERING, July 1943, page 464



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